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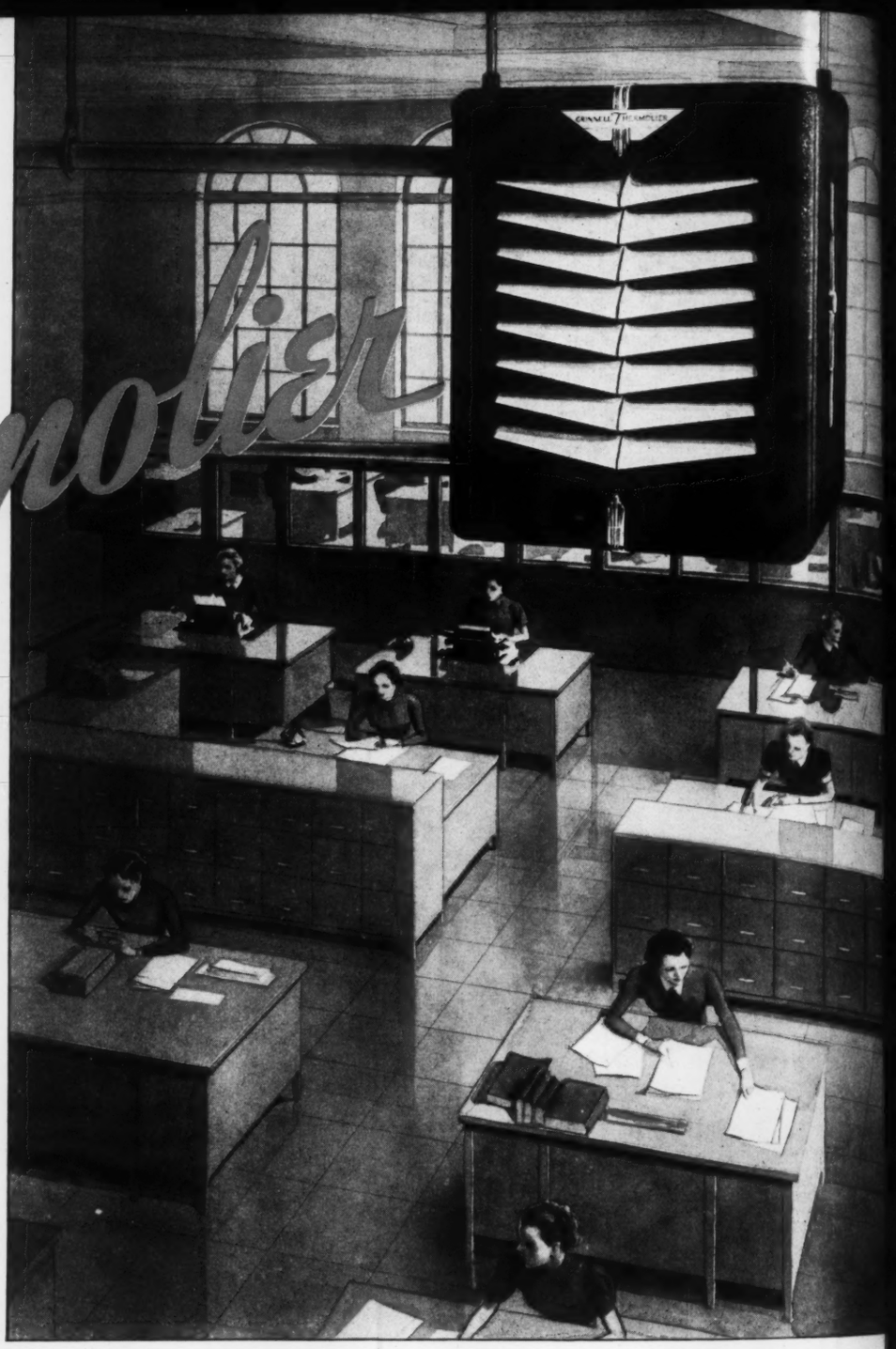
Heating · Piping and Air Conditioning

Two of this month's authors take us travelling, one by Pan-American Clipper to study the air conditioning problems of a tropical climate, the other to investigate American air conditioning practice through the eyes of an English visitor

1937's
UNIT HEATING NEWS

Thermolier
DE LUXE TYPE



de luxe **HOUSING . . .**

created by a leading industrial designer in crackle black and satin chrome . . . modern design . . .

de luxe **PERFORMANCE**

resulting from the time-tested Thermolier heating unit, plus new louvres tilted at the exact angle for widespread distribution of heat.

• • •

Now a Thermolier de luxe type! The proved performance record of Factory and Industrial Type Thermoliers, *plus* appearance worthy of showroom, lobby, or office, *plus* better distribution of heat!

Every one of Thermolier's famous 14 Points of Superiority is built into this new Thermolier. The exclusive Internal Cooling Leg that constantly removes "lazy water" . . . the correctly pitched, expansion-bend tubes . . . the leak-proof joints . . . and eleven others.

The specially-designed motor and fan are completely concealed. Inlet and outlet pipes are at the rear. And Thermolier de luxe type carries on Thermolier's reputation as a miser of fuel.

You should know all about this new Thermolier. Write today for full particulars, without obligation. Grinnell Company, Inc., Executive Offices, Providence, R. I. Branch offices in principal cities.

18 MONTHS' RESEARCH ASSURES SATISFACTORY PERFORMANCE!

Consistent with the policy of Grinnell, a year and a half of intensive research was carried out on Thermolier de luxe type before its announcement. It is offered to you thoroughly tested.

GRINNELL THERMOLIER

THE UNIT HEATER WITH 14 POINTS OF SUPERIORITY

The Editor's Page

■ **New York City** leads the cities in the United States—and the world too—in the size of its air conditioning load, Chicago comes second, and Washington, D. C., third, according to a recently-completed survey of air conditioning made by the department of technical development and research of Consolidated Edison Co. of New York. As of the beginning of the year, the horsepower figures are New York, 84,500; Chicago, 58,500; Washington, 39,500 (25,000 for government buildings). . . . Though first in the size of its air conditioning load New York rates second to Chicago in number of installations—New York's four boroughs surveyed—Manhattan, the Bronx, Brooklyn and Queens, having 1300 at the beginning of 1937 and Chicago 1370.

■ **The 20,000th** commercial "Freon" compressor was completed by York Ice Machinery Corp. last month. . . . Because more than 80 per cent of its products are known by the trade name "American-Marsh," and because over 50 per cent of them are not steam operated, the American Steam Pump Co., Battle Creek, Mich., has changed its name to American-Marsh Pumps, Inc. There have been no changes in financial structure, management, or address .... Warren Webster & Co., Camden, N. J., has become exclusive national distributors for the unit heaters manufactured by John J. Nesbitt, Inc., of Holmesburg, Philadelphia, Pa. The arrangement does not affect the existing marketing plans for Nesbitt's other products. . . . Lincoln Electric Co. is building a 200,000 sq ft all welded rigid frame addition to its plant at Cleveland. A two story building, it employs the new welded rigid frame saw tooth design utilizing "tree form" columns. A total of 29,600 linear feet of electric welding using over 5½ tons of welding electrode will be required.

■ **An increase in passenger revenues** of 32 per cent for May over the corresponding period last year is attributed by the Seaboard Air Line Railway in great measure to the shortening of schedules and the fact that all coaches and pullmans on the railroad's through trains are completely air conditioned. . . . The various types of air conditioned cars in service on the Seaboard include 17 passenger and baggage cars, 41 coaches, 22 dining cars, 45 sleeping cars, and 11 lounge cars. . . . The crack express train covering 376 miles between Tokyo and Kobe, Japan, named the "Tsubame" or "Chimney Swallow," will add to the comfort of its patrons by using air conditioned diners, according to a report from the machinery division of the Department of Commerce by Carl H. Boehringer, assistant trade commissioner at Tokyo. . . . Out of 1587 dining cars owned or operated by U. S. railroads, 985 are now air conditioned and 28 new diners with this feature are now on order, the machinery division remarks.

■ **Increasing interest** has been shown in air conditioning in the Netherland Indies during the past six months, according to "World Machinery News" issued by the Bureau of Foreign and Domestic Commerce. Due to excessive heat there is a definite need of air conditioned railway coaches and there is little doubt that a potential market exists for this type of equipment. . . . The same source says a new hospital with accommodations for 1000 patients to be erected in the Straits Settlements has been approved. An innovation will be air conditioning in upper class wards.

■ **Nearly $20,000,000** will be spent this year for non-domestic condensing units, according to A. H. Reinach of Nash-Kelvinator Corp., because of the rapid development of air conditioning and the expansion of the market for commercial refrigeration. . . . According to J. K. Knighton, sales manager of commercial air conditioning for Kelvinator Division, Nash-Kelvinator Corp., 1937 air conditioning sales will double the total of last year. . . . Evidence of what air conditioning will do to increase restaurant business is shown by a business increase of 25 per cent in Miss Morris' Eat Shop in Minneapolis, reports Westinghouse Electric and Mfg. Co. The net profit from a 25 per cent increase in business will amount to at least $2500 before deducting the $765 expense for air conditioning or a net of $1735 per year. The system was purchased on a 36 month F.H.A. contract. By the time it is paid for, with three years operating expense added, the entire cost at that time will be $4200. The added profits due to its installation will be at least $7500, leaving a credit balance of $1100 per year for the first three years and $2200 per year thereafter.

■ **Installed cost** of equipment sold by the members of the Air Conditioning Manufacturers' Association in July totalled $7,508,778, an increase of 52.1 per cent over the total reported for the corresponding month in 1936. . . . July sales were greater than for any month last year and brought this year's total to $68,306,175, which is 37 per cent higher than the volume for all 1936.

■ **There is no end** to what can be air conditioned these days, if one believes everything he reads. . . . The latest is a carbonated beverage, which—according to the sponsor—"air conditions your insides by the expansion of the cool carbonic gas."

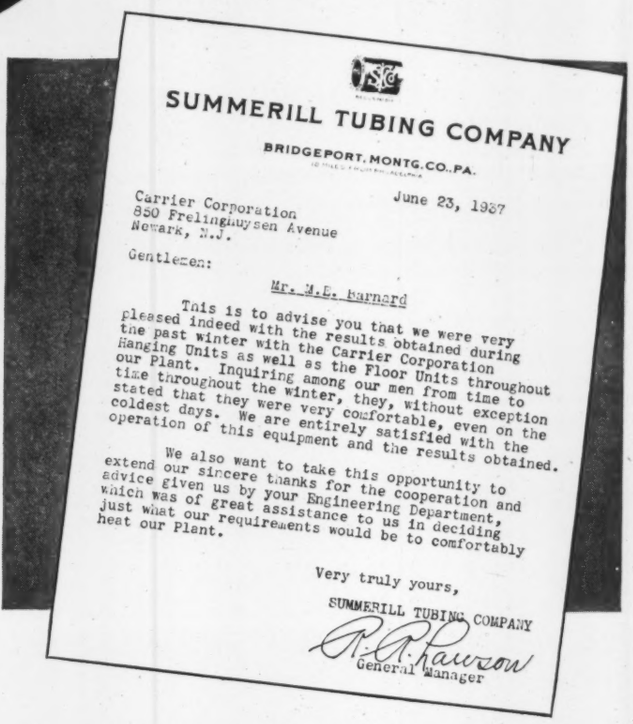
■ **His Majesty, Farouk the First,** was kept comfortably cool by American science while he was being invested as King of Egypt. . . . He took oath in the air conditioned assembly chamber of Parliament Hall, Cairo. Peter Gray, engineer for Carrier Corp., was present during the ceremony, checking the operation of the newly installed air conditioning apparatus. . . . Temperature conditions equal to those in the famous Caves of Roquefort are maintained by air conditioning in the exhibition pavillion of the Roquefort cheese manufacturers at the Paris Exposition, according to Carrier Corp. . . . According to Airtemp, Inc., the duct work for the new year 'round air conditioning system for the Madison Square Garden arena, New York City, is more than a mile in length and over 300,000 lb of sheet metal went into its construction. . . . The entire system is subject to remote control from the engineer's office where there are located recording instruments to show the temperatures at various points in the arena, and fan and damper controls.

■ **Because of the rapid** industrial expansion in the South, the International Acetylene Association will hold its annual convention for the first time south of the Mason and Dixon line. The dates are November 10, 11, and 12, and the place, Birmingham, Ala. . . . Extensive changes in the American Standard safety requirements for the installation of air conditioning and ventilating systems, have resulted in the submitting of a revision of the standard code to the American Standards Association for approval, according to "Industrial Standardization."

STEAM COSTS CUT by Replacing Pipe Coils with Carrier Unit Heating

Summerill Tubing Co. Praises Economy of Carrier Equipment in Bridgeport, Pa. Plant

1. HERE IT IS . . . the letter written by General Manager Lawson. If you're interested in heating efficiency *plus* low operating cost—read every word.



SUMMERILL TUBING COMPANY

BRIDGEPORT, MONTG. CO., PA.

June 23, 1937

Carrier Corporation
850 Frelinghuysen Avenue
Newark, N.J.

Gentlemen:

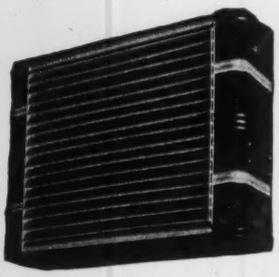
Mr. M.E. Barnard

This is to advise you that we were very pleased indeed with the results obtained during the past winter with the Carrier Corporation Hanging Units as well as the Floor Units throughout our Plant. Inquiring among our men from time to time throughout the winter, they, without exception stated that they were very comfortable, even on the coldest days. We are entirely satisfied with the operation of this equipment and the results obtained.

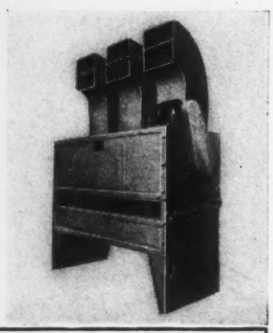
We also want to take this opportunity to extend our sincere thanks for the cooperation and advice given us by your Engineering Department, which was of great assistance to us in deciding just what our requirements would be to comfortably heat our Plant.

Very truly yours,
SUMMERILL TUBING COMPANY
A.R. Lawson
General Manager

2. HARD TO HEAT! With direct radiation, heat rises to the roof. The efficient Kroy units direct the heat *downward*, exactly where needed, and, as Mr. Lawson reports, "our men . . . without exception . . . were comfortable on coldest days."



3. UNUSUAL FEATURE of the Carrier Kroy unit is the return bend construction of the non-ferrous coil, which eliminates expansion and contraction strains, and is guaranteed to withstand working pressures of 200 lbs. per sq. in. Note attractive design.



4. FLOOR MOUNTED UNITS like this were also selected by Summerill. Reasons: High velocity of heated air—quiet operation—attractive design—small space required—and maximum operating economy. Thermajust control makes heating automatic—prevents wasteful over heating.

Take a Tip from Summerill . . .
Install Carrier Unit Heating—NOW!

● How much was your heating bill last winter? Scores of companies have reduced fuel costs as much as 35%, by replacing direct radiation with Carrier Unit Heating—to say nothing of the increased efficiency and increased comfort of employees. And it's a *permanent* saving—one you enjoy year after year for countless years to come.

Look at the record: Thousands of Carrier Unit Heaters installed 10, 12, even 15 years ago, are still on the job—operating as economically, as efficiently as the day they were made. That's proof that Carrier Unit Heaters are built to endure! Consider construction. All coils are non-ferrous and all connections strongly brazed. Return bend coil construction prevents strain due to expansion and contraction, eliminates leakage. Newly-designed fans reduce required horsepower as much as 50%. Balanced assemblies prevent noise and vibration. No wonder Carrier Unit Heaters are selected by the industries of the world!

What can Carrier Unit Heating do for you? Your local Carrier dealer will tell you, quickly and without obligation. Call him today—or mail coupon.

CARRIER CORPORATION, Desk 903, Syracuse, N. Y.
Please send me complete information on Carrier Unit Heating

for .....
(TYPE OF BUILDING)

Name .....
Company .....
Address .....
City .....

Heating · Piping and Air Conditioning

with which is merged the "Aerologist"

Design, Installation
Operation, Maintenance
in Industry and Large Buildings

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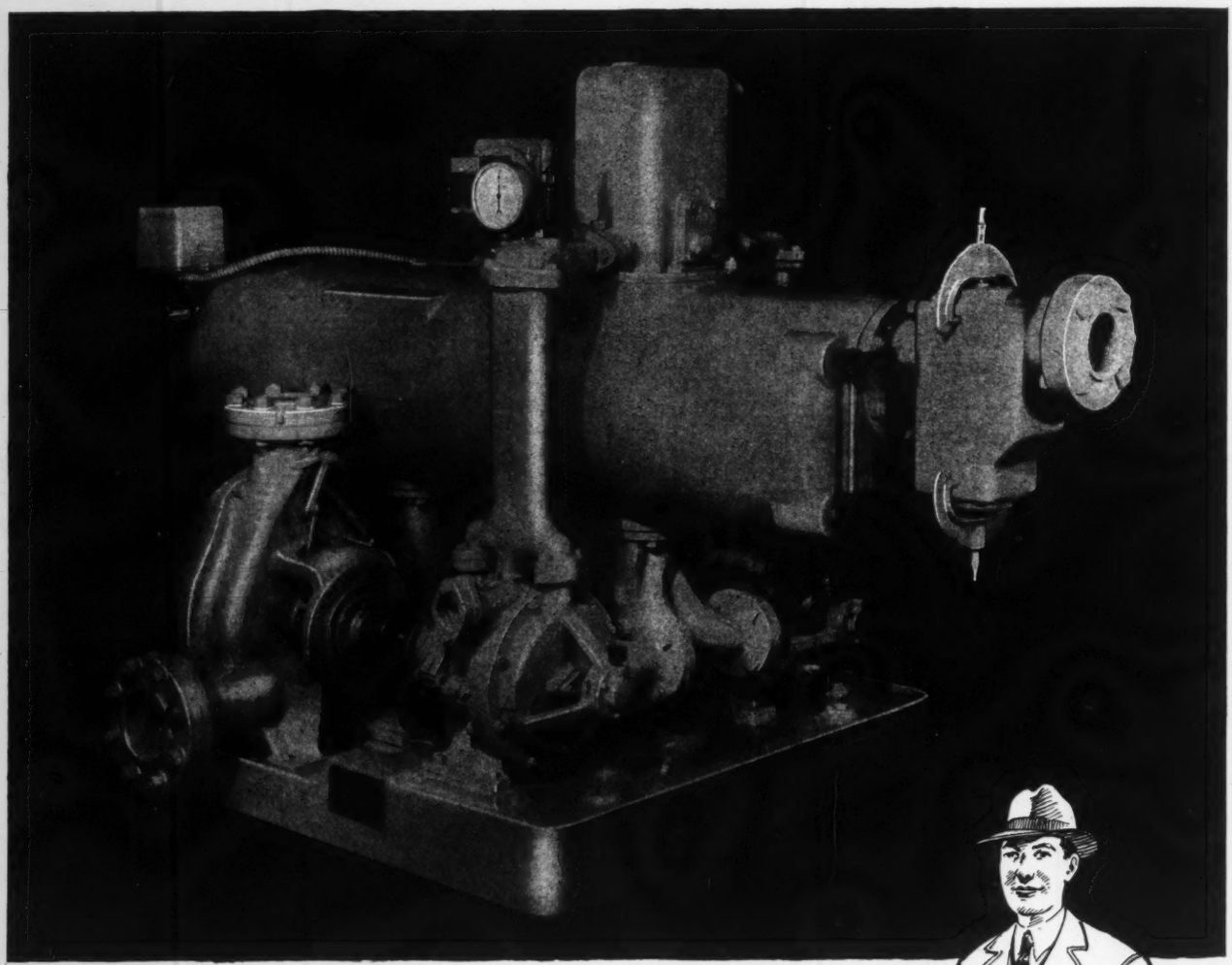
THIS ECONOMICAL HEATING PUMP OPERATES WITHOUT ELECTRIC CURRENT



The Jennings Vapor Turbine Return Line Vacuum Heating Pump requires no electric current, eliminating the one biggest expense item in the operation of an ordinary return line pump. This is because the motive power is a special turbine operating on steam direct from the heating system. Steam used to drive the turbine is returned to the system for heating, with little heat loss.

The really important saving promoted by the Jennings Vapor Turbine is in the system itself, however, for this is the only heating pump that can operate continuously with economy. Continuous operation means uniform circulation, and uniform circulation means a big saving in steam.

This pump has but one moving part, no internal wearing parts, and no internal lubrication. It is compact and quiet, and will give long and satisfactory service at the lowest maintenance cost. Bulletin No. 246 tells all about it. It is free on request.

THE NASH ENGINEERING COMPANY

SOUTH NORWALK, CONNECTICUT, U. S. A.

Heating · Piping and Air Conditioning

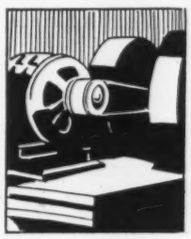
Vol. 9, No. 9 • *September, 1937*

New Data

Aids Refrigeration Piping Design

"Freon" Suction Line Piping

AS was pointed out in an earlier issue,[1] in the design of a refrigeration plant it is essential that an evaporator and a compressor of suitable capacity and balance be chosen to handle the load, but it is even more vital that the piping for the liquid and suction lines between this equipment be properly planned and installed. Data were given in the first article on liquid line piping for the refrigerant "Freon-12" ("F-12" for short) and curves of pressure drop for the liquid through thermal expansion valves and solenoid shut-off valves were presented. Our subject this month is the suction piping for "Freon-12."



The saturated vapor of "Freon-12" is very heavy, having a specific volume of 1.637 cu ft per lb at 0 F and 0.780 at 41 F; or about five to six times as heavy as ammonia vapor at the same temperatures. Inasmuch as the refrigerating effect is relatively low, large volumes must be circulated and this, coupled with "Freon's" heavy weight, accentuates the need of velocity control to avoid large pressure drops.

Figs. 1 and 2 show pressure drops of "Freon-12" vapor for various tonnages through 100 lineal feet of copper pipe of different sizes for saturated gas at 0 F. Figs. 3 and 4 show such pressure drops for saturated gas at 41 F. Table 1 indicates the pressure drop caused by valves and fittings in terms of the equivalent lineal feet of pipe.

None of the values of Table 1 is "guaranteed," but the data are derived from the best information available and are presented simply as a guide to the designing engineer.

It is considered good design for the pressure drop in the suction line from the farthest distant evaporator to the compressor not to exceed 3 or 4 lb per sq in. The drop in pressure of the suction gas passing through the compressor suction valve into the crankcase may be as much as ½ or 1 lb which, with the above mentioned limits, will total 4 to 5 lb differential between the evaporator and compressor; this should be considered the maximum allowable for any condition. If the drop in pressure is greater than this optimum, the compressor suffers in capacity by having to run at too low a back pressure. No doubt this point accounts for the failure of some compressor installations to produce their rated tonnage whereas, the compressor may not be at fault nearly as much as the piping connections.

For example, assume we have a layout for air conditioning at 41 F suction temperature in which there is a load of 12 tons of refrigeration with a 50 ft length of suction line containing two valves and five elbows:

Assume a 1⅝ in. suction line—

Length of suction line	50 ft
Equivalent length for 2 valves	16 ft
Equivalent length for 5 elbows	35 ft
Total	101 ft.

This pressure drop is 2.7 lb per sq in., which is quite satisfactory. However, if the line is 150 instead of 50 ft long with two valves, four tees and ten elbows, the pressure drop is much larger:

Length of suction line	150 ft
Equivalent length for 2 valves	16 ft
Equivalent length for 4 tees	40 ft
Equivalent length for 10 elbows	70 ft
Total	276 ft

Table 1—Pressure Drop Due to Valves and Fittings in Equivalent Length of Pipe for "Freon-12"

Copper Pipe Sizes..In. (O.D.)	⅝	⅞	1⅛	1⅜	1⅝	2⅛	2⅝	3⅛	3⅝	4⅛	5⅛	6⅛	8⅛	10¾	12¾
Elbow	4	5	5	6	7	8	10	12	15	18	24	30	40	50	60
Tee	5	6	7	8	10	12	15	18	20	25	30	40	50	60	75
Valve	5	6	6	7	8	10	12	20	25	30	40	50	60	80	100

By R. C. Doremus*

Basic data on the design and installation of piping for industrial and commercial refrigeration work and air conditioning are presented. As a refrigeration engineer, Mr. Doremus has had the opportunity to observe jobs which have been poorly designed, has concluded that lack of information is the cause. He therefore makes available his data in this form

*Chief Engineer, Detroit Ice Machine Co., Member of Board of Consulting and Contributing Editors.

[1] "New Data for Refrigeration Piping Design," by R. C. Doremus, Heating, Piping and Air Conditioning, August, 1937, pp. 492-495.

FIG. 1
PRESSURE DROP IN SUCTION LINES OF "FREON-12" @ 0 F SUCTION TEMPERATURE, PER 100 FT LENGTH OF PIPE

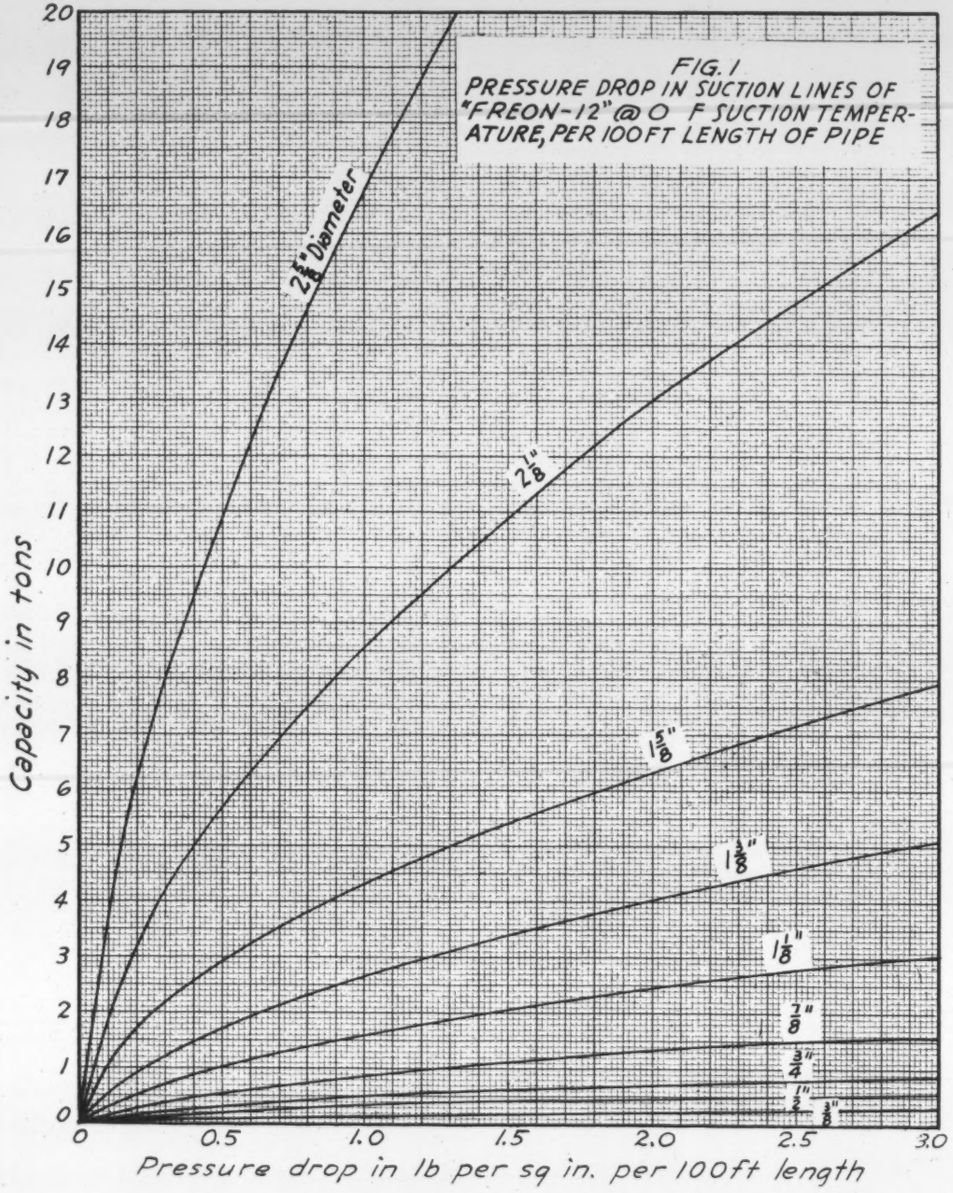
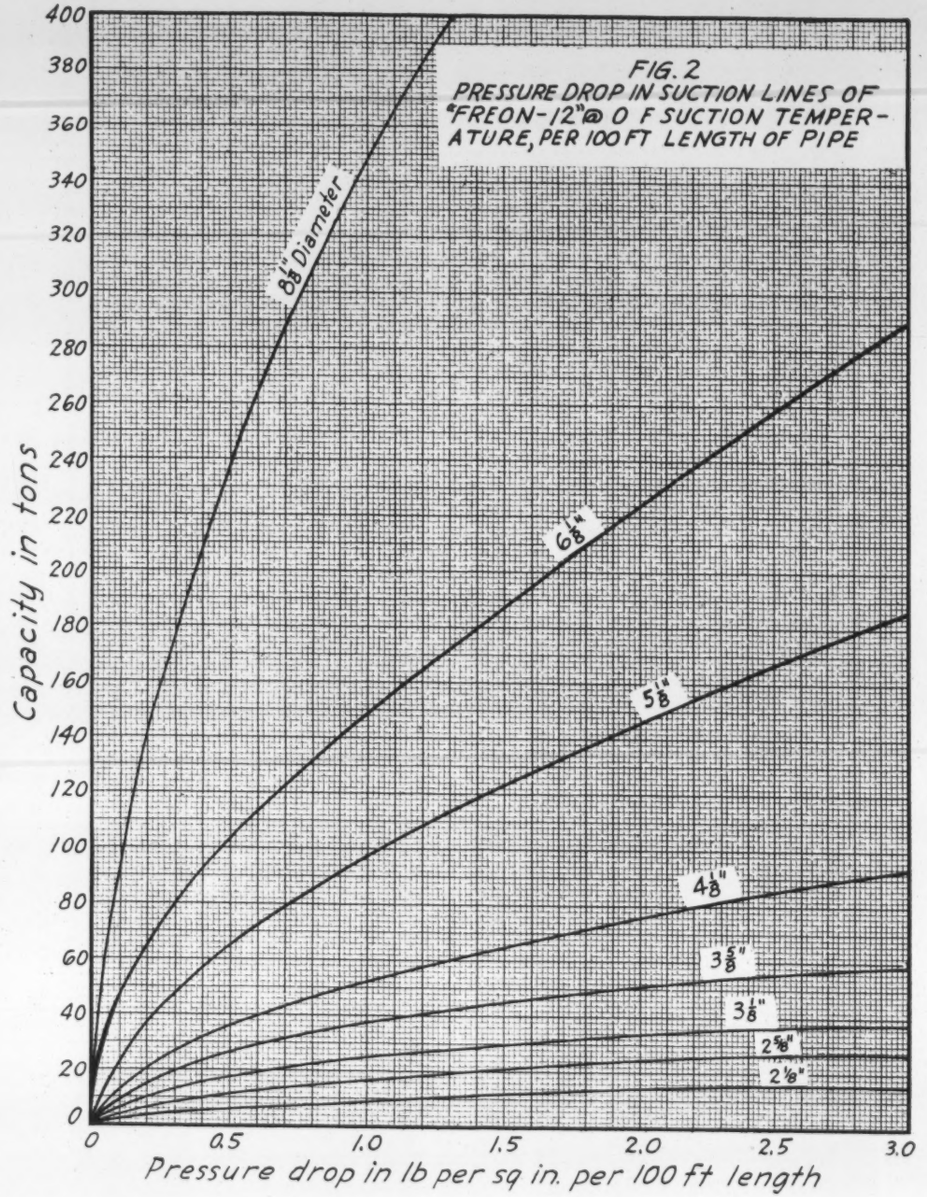


FIG. 2
PRESSURE DROP IN SUCTION LINES OF "FREON-12" @ 0 F SUCTION TEMPERATURE, PER 100 FT LENGTH OF PIPE



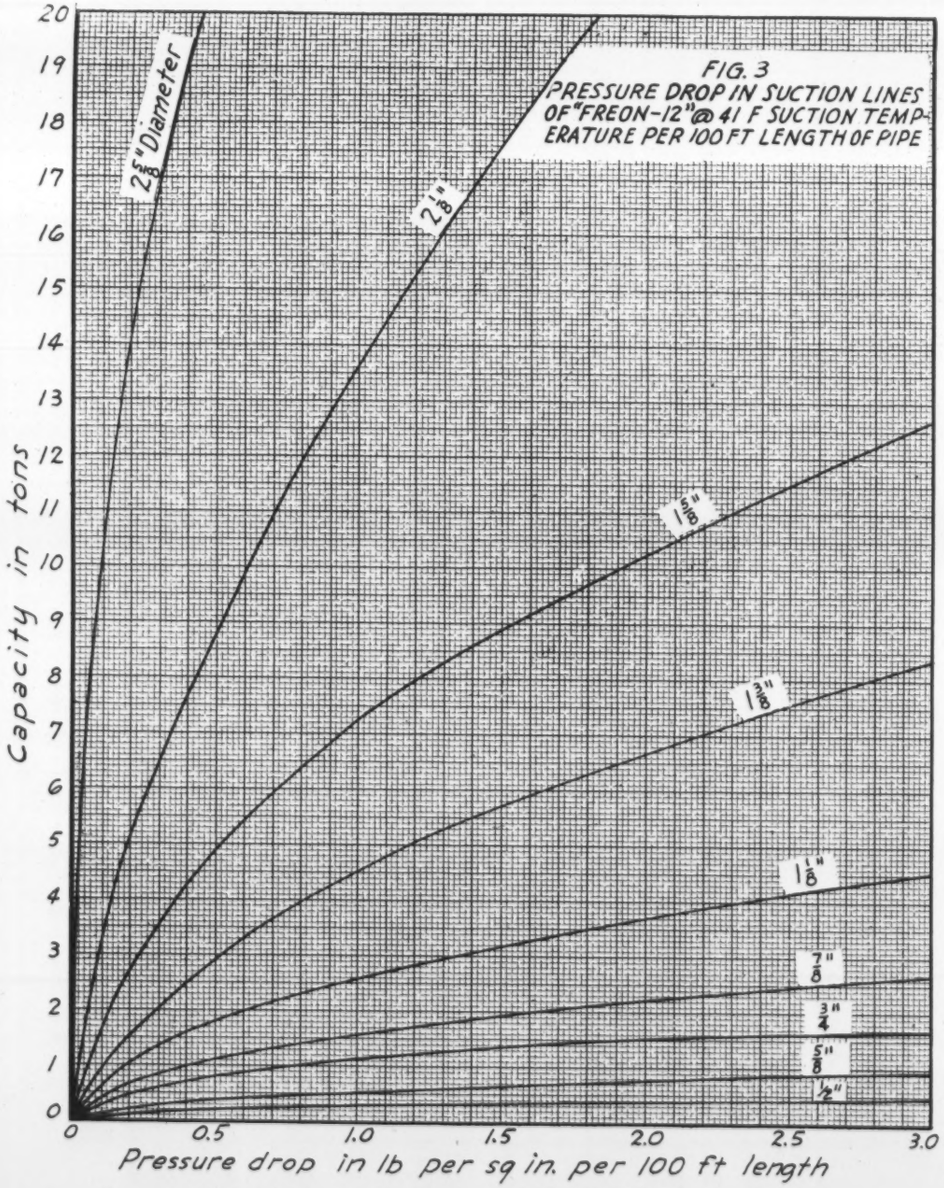


FIG. 3
PRESSURE DROP IN SUCTION LINES OF "FREON-12" @ 41 F SUCTION TEMPERATURE PER 100 FT LENGTH OF PIPE
2 5/8" Diameter
2 1/8"
1 3/8"
1 1/8"
7/8"
3/4"
5/8"
1/2"
Capacity in tons
Pressure drop in lb per sq in. per 100 ft length
0
0.5
1.0
1.5
2.0
2.5
3.0
0
1
2
3
4
5
6
7
8
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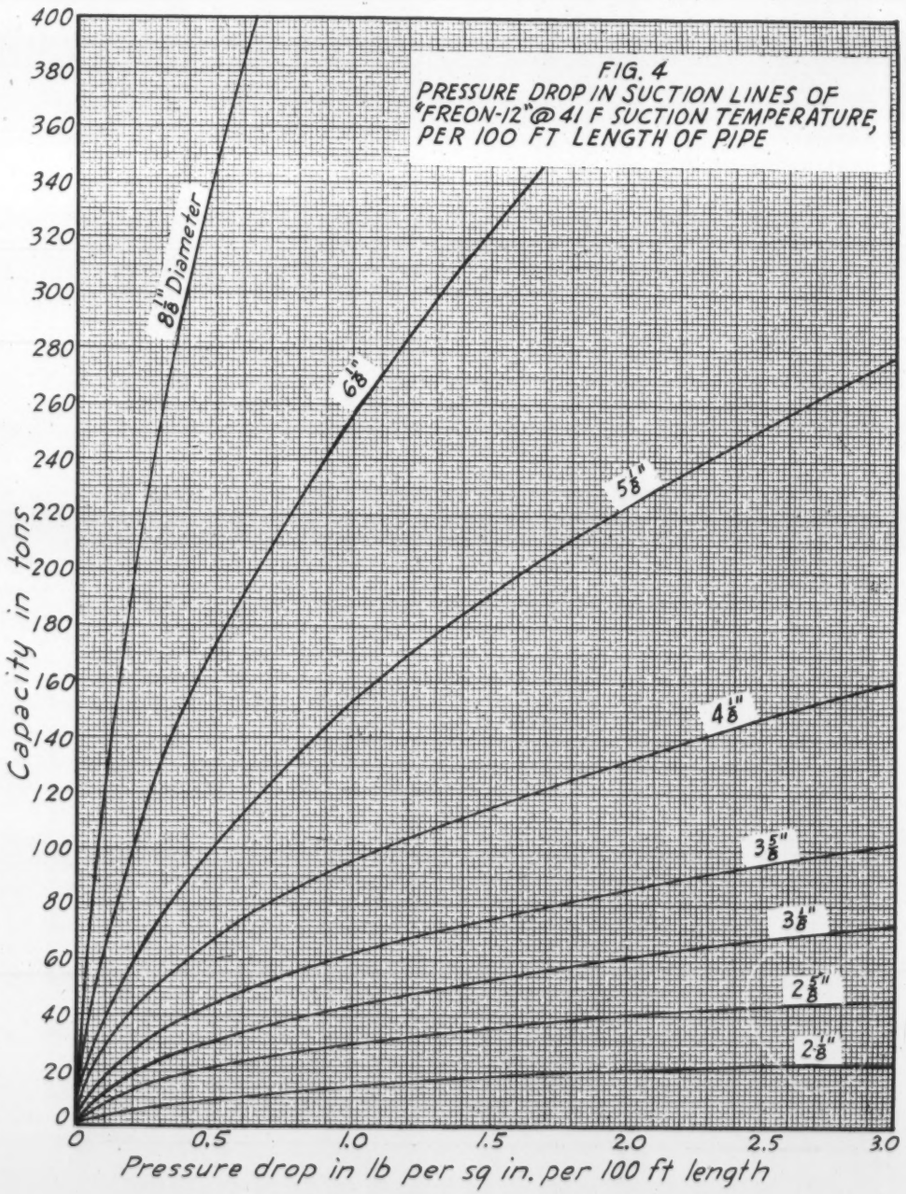


FIG. 4
PRESSURE DROP IN SUCTION LINES OF "FREON-12" @ 41 F SUCTION TEMPERATURE, PER 100 FT LENGTH OF PIPE
8 1/8" Diameter
6 1/8"
5 1/8"
4 1/8"
3 5/8"
3 1/8"
2 5/8"
2 1/8"
Capacity in tons
Pressure drop in lb per sq in. per 100 ft length
0
0.5
1.0
1.5
2.0
2.5
3.0
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
300
320
340
360
380
400

Table 2—Velocity Chart for "Freon-12" Suction Vapor at Various Suction Temperatures for Copper Piping of Different Diameters, in Feet per Minute per Ton of Refrigeration

Suct. Temp. F	Corr. Pressure Lb/Sq In.		Sp. Vol. of Sat. Vapor Cu Ft/Lb	Heat of Sat. Vapor Btu/Lb	Heat of Liquid @94F Btu/Lb	Refrig. Effect Btu/Lb	Weight Lb/Min/Ton	Actual Displacement Cu In./Min/Ton	Copper Pipe Sizes, In. O.D.														
	Abs.	Ga.							½	⅝	⅞	1⅛	1⅜	1⅝	2⅛	2⅝	3⅛	3⅝	4⅛	5⅛	6⅛	8⅛	10⅛	12⅛
−40	9.32	11.0*	3.911	73.50	29.30	44.20	4.528	30.600	17.460	10.930	5.250	3.040	2.012	1.562	823	534	374	276	212	136	94.8	54.2	36.5	25.3
−35	10.67	8.2*	3.500	74.10	29.30	44.80	4.460	27.050	15.420	9.660	4.648	2.685	1.777	1.382	726	472	351	246	187	120	93.9	48.0	32.2	22.4
−30	12.02	5.4*	3.088	74.70	29.30	45.40	4.410	24.500	13.990	8.750	4.201	2.432	1.610	1.250	659	427	300	221	170	109	76.0	43.3	29.2	20.2
−25	13.65	2.1*	2.780	75.20	29.30	45.90	4.360	20.920	11.940	7.465	3.585	2.075	1.375	1.038	562	364	256	189	145	93.1	65.0	37.1	24.9	17.3
−20	15.28	0.6	2.474	75.87	29.30	46.57	4.300	18.400	10.500	6.570	3.161	1.828	1.210	940	495	321	224	166	127	82.0	57.1	32.6	21.9	15.2
−15	17.24	2.5	2.240	76.47	29.30	47.17	4.245	16.450	9.380	5.875	2.822	1.635	1.080	840	442	289	201	148	114	73.2	51.0	29.1	19.6	13.6
−10	19.20	4.5	2.003	77.05	29.30	47.75	4.185	14.450	8.250	5.160	2.480	1.436	950	738	389	252	177	130	100	64.4	44.8	25.5	17.2	12.0
−5	21.53	6.8	1.873	77.60	29.30	48.30	4.140	13.050	7.460	4.660	2.238	1.296	858	667	351	228	159	117	90.4	58.1	40.5	23.1	15.5	10.8
0	23.87	9.2	1.637	78.21	29.30	48.91	4.085	11.530	6.580	4.120	1.980	1.146	758	590	310	202	141	104	80.0	51.4	35.8	20.5	13.7	9.5
+5	26.50	11.8	1.480	78.80	29.30	49.50	4.045	10.330	5.900	3.695	1.775	1.026	680	528	278	180	126	93.3	71.5	46.0	32.0	18.3	12.3	8.5
+10	29.37	14.6	1.351	79.30	29.30	50.00	4.000	9.330	5.325	3.335	1.600	926	613	477	251	163	114	84.1	64.8	41.6	29.0	16.5	11.1	7.7
+15	32.57	17.8	1.236	79.90	29.30	50.60	3.952	8.425	4.810	3.010	1.446	836	554	430	226	147	103	76.0	58.4	37.5	26.1	14.9	10.0	7.0
+20	35.75	21.0	1.121	80.49	29.30	51.19	3.906	7.560	4.315	2.700	1.300	752	498	386	203	132	92.5	69.0	52.5	33.7	23.5	13.4	9.0	6.2
+25	39.45	24.7	1.031	81.05	29.30	51.75	3.870	6.890	3.930	2.460	1.182	684	453	352	185	120	84.1	62.1	47.8	30.7	21.4	12.2	8.3	5.7
+30	43.16	28.4	0.939	81.61	29.30	52.31	3.828	6.205	3.542	2.220	1.066	617	408	317	167	108	75.9	56.0	43.1	27.6	19.3	11.0	7.4	5.1
+35	47.52	32.8	0.866	82.15	29.30	52.85	3.782	5.660	3.230	2.022	972	562	372	289	152	98.8	69.0	51.0	39.2	25.2	17.5	10.0	6.7	4.7
+40	51.68	37.0	0.792	82.71	29.30	53.41	3.750	5.130	2.930	1.835	880	510	338	262	138	90.0	62.7	46.3	35.6	22.9	15.9	9.1	6.1	4.2
+41	52.60	37.9	0.780	82.81	29.30	53.50	3.740	5.030	2.875	1.800	865	500	324	257	135	88.4	61.5	45.3	35.0	22.4	15.6	8.9	6.0	4.1
Maximum Recommended Velocity, fpm									450	500	700	900	1,000	1,100	1,250	1,400	1,600	1,800	2,100	2,500	2,800	3,200	4,000	4,600

*All pressures marked (*) are vacuum expressed in inches of mercury.

Example: The velocity caused by 1 ton through a 5⅛ in. pipe @ 41 F suction temperature is 22.4 fpm. The maximum tonnage that should be carried in this line is 2500 ÷ 22.4 = 113 tons of refrigeration.

The pressure drop in this case is 7½ lb per sq in., which is too large. It would be far better to use a 2⅛ in. line, in which case the pressure drop would be computed as follows:

Length of suction line	150 ft
Equivalent length for 2 valves	20 ft
Equivalent length for 4 tees	48 ft
Equivalent length for 10 elbows	80 ft
Total	298 ft

In this case the pressure drop would be only 2.4 lb per sq in., which is very satisfactory for good design.

Let us assume, however, that the suction temperature is 0 F. The pressure drop would then be 5¼ lb per sq in., and at this low back pressure, it would be better to use a 2⅝ in. suction line in which the pressure drop would be less than 2 lb. per sq in.

Table 2 is a table of gas velocities of "Freon-12" suction vapor in feet per minute per ton in various sizes of copper pipe for saturated temperatures from —40 to +41 F. This shows velocity variations according to suction pressure or temperature, which are greatly affected by the specific volume of the vapor and the actual displacement of the vapor per ton under these conditions. It enables the designer to ascertain the actual velocity of the suction vapor in various suction pipe sizes at any suction pressure or temperature by simply multiplying these corresponding values in the table by the number of tons of refrigeration to be carried in the pipe.

The maximum velocity for good design is shown at the bottom of the table for each pipe size so that each branch or main of the suction line may be properly proportioned in size.

Assume we have a load of 60 tons to be handled with 41 F gas. Assume a 4⅛ in. pipe size:

60 × 35 (from Table 2) = a velocity of 2100 fpm

Inasmuch as the maximum for this pipe size is 2100 fpm, a 4⅛ in. suction main will be satisfactory.

If the load is 80 tons of refrigeration, 80 × 35 gives a velocity of 2800 fpm which is too high a velocity for a 4⅛ in. pipe and it would be better to use a 5⅛ in. line. 80 × 22.4 gives a gas velocity of 1792 fpm and this is well within the maximum of 2500 for this size line.

On the other hand, let us assume that the 60 ton load mentioned above were at 0 F instead of at 41 F. This rarer gas will cause a higher velocity and the 4⅛ in. line which was satisfactory for the 60 ton load at 41 F is now too small:

60 × 80 gives a velocity of 4800 fpm, whereas 2100 is maximum.

Also, a 5⅛ in. line is too small for this condition:

60 × 51.4 gives a velocity of 3084 fpm, whereas 2500 is maximum.

It would be better design to use a 6⅛ in. pipe:

60 × 35.8 gives a velocity of 2148 fpm, which is well within the maximum of 2800 fpm.

Most systems operate with 20 to 30 degrees superheat, which will increase the velocities by approximately 10 percent; however, this has been taken into consideration in establishing the maximum velocities.

It is safe design to plan piping for suction lines in accordance with these data; it will be noted that these maximum velocities vary from 450 to 4600 fpm according to the pipe size.

The back pressure will determine the number of tons that may be carried in any size pipe. As with other refrigerants, the discharge line may be the same size as the suction or one size smaller without any detriment inasmuch as the gas is compressed to a smaller volume giving lower velocities than in the suction line and the length of this line from the compressor to the condenser is

[*Concluded on page 547*]

Leaving New York, I watched with awe the "Mountains of Manhattan" recede

As Others See Us .....

By John Pryke

.....an Englishman Studies American Air Conditioning

LAST October I set sail from London for America to study air conditioning in the United States, in view of the enormous strides that have been made in that particular field during the past few years.

Perhaps, before starting a description of one's travels, a few words of explanation as to the circumstances leading to them may be in order. The London County Council—the municipal governing body of London—awards annually two engineering fellowships, the purpose of which is to enable young men who have had an engineering training plus practical business experience to study their particular line of work abroad in order to improve their knowledge and experience. The fellowships are awarded for any particular branch of engineering for which a candidate should care to apply.

As I had been engaged in heating and air conditioning work in London for a number of years it was felt that a study of air conditioning in the United States would be helpful and instructive, so an application for one of the fellowships was made, with successful results.

The journey over was made via Quebec and Montreal and New York was reached on October 25. First impressions are always supposed to be of vital importance and in this connection two things stood out at once—size and noise.

Two months were spent in New York with a fortnight's trip to Boston interposed. During this period such installations as those at Rockefeller Center, the Metropolitan Life Insurance Co., the Holland Tunnel, the Roxy cinema, and the Beech Nut chewing gum factory were inspected and much valuable information was gained. Of course comfort cooling could not be studied in actual operation, but types of plant and equipment were seen, and in addition, the larger field of general American building construction was studied to a certain extent. This also was a period of acclimatization, during which I got used to America and Americans and a little of my British insularity began to wear off.

During the Boston trip a cotton spinning mill, a printing works and a brewery were inspected and a visit was paid to the Massachusetts Institute of Technology where I was impressed by the completeness of the instructional facilities available—especially as regards the practical instruction in laboratories.

On January 4 the main trip through the U. S. A. was started, and en route to Hoboken I gazed with suitable awe at the "Mountains of Manhattan" receding across the sparkling waters of the sunlit Hudson River. Philadelphia was the first port of call and here, amongst other things, a reversed refrigeration cycle installation and a theater with high speed nozzle air inlet were seen. Also several pleasant and informative discussions took place on radiant heating and high speed air discharge work.

From Philadelphia to Washington was a short step and here in the nation's Capital I saw much of great interest. The new government buildings have practically all been air conditioned either in their entirety or partially, and the Capitol, Senate Office Building, Justice

Building, Supreme Court, Archives Building, Congressional Library Annex, and Department of Interior Building were well worth inspection. Probably some of the largest air conditioning installations now projected in the U. S. A. are these in Washington, and certainly the acute dehumidification problem encountered during the summer months justifies the expenditure involved.

The largest project is of course that involving the Capitol, Senate Office Building and the two Representatives' Office Buildings. These four structures are to be conditioned entirely, chilled water being supplied from a common refrigerating plant of about 5000 tons capacity, situated in the existing district steam generating station about 1½ miles away. In all the government buildings seen I was struck by the high standard of the installations and by the comprehensive control panels provided in the majority of cases.

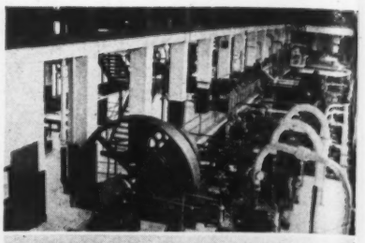
The next city visited was Pittsburgh, where I got the impression that more rain fell during my visit than could have ever happened before. The *A.S.H.V.E.* laboratory was inspected, and in Mr. Houghten's absence, Mr. Gutberlet very kindly acted as guide and explained the various research projects being carried out.

At Cincinnati, next on the itinerary, Mother Nature again took a hand and not content with just rain, as at Pittsburgh, proceeded to pile up the Ohio River to six times its usual height, and this, while providing an awesome and not easily forgotten spectacle, did not exactly help my air conditioning researches, as naturally, if people are on emergency duty 24 hours per day they cannot be expected to devote much time to the visitor.

The annual meeting of the *A.S.H.V.E.* was of course the most important event in St. Louis, and here I had my first glimpse of the American combining business and pleasure—apparently to his own and everyone else's satisfaction. The meeting finished on a Wednesday, and I stopped over in St. Louis until the end of the week. Amongst various buildings and installations inspected the Statler Hotel furnished an interesting example of a form of air conditioning applied to an existing building with the minimum of alteration, whilst the Municipal Auditorium demonstrated air conditioning en masse and the growth of steam jet refrigeration, a 1200 ton capacity machine being installed. The most interesting thing seen in this district, however, was the central ammonia refrigerating installation of the St. Louis Refrigeration and Cold Storage Co. At present one main and one substation are in use, serving a network of ammonia gas and liquid lines running under the streets, a total of about 32 miles of piping being installed. A great number of buildings are served, each having its own evaporator which provides chilled water for the air conditioning equipment. The machines already installed have a total capacity of about 3400 tons and owing to the ever increasing demand large extensions are contemplated. The ammonia compressors used are of the horizontal, two stage type driven by synchronous motors.

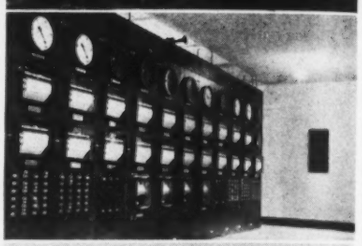
From St. Louis a long hop south was made to New Orleans, and this journey gave me my first glimpse of the South. In New Orleans it seemed that air conditioning had not progressed to the same extent as in the North and East. In the city itself, a dehumidification problem exists, and the principal reason for the slowness of progress appears to be financial rather than technical. In the surrounding country where cool nights are experienced the attic fan for night operation for the smaller private house appears to be the rule.

An interesting reversed refrigeration installation was inspected in the Louisiana Light and Power Co. offices at Algiers, just across the river from New Orleans, with

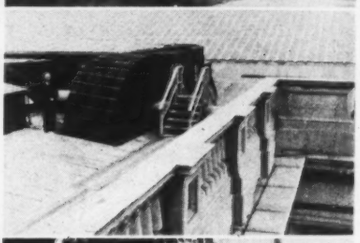


Engineering laboratory at M. I. T. The completeness of the instructional facilities impressed me .....

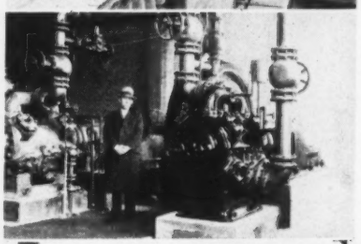
Some of the largest air conditioning installations are in Washington, D. C. The Federal Triangle is an air conditioned one.....



I was struck by the comprehensive control boards provided for most of the Washington installations. This one is in the Interior department's building .....



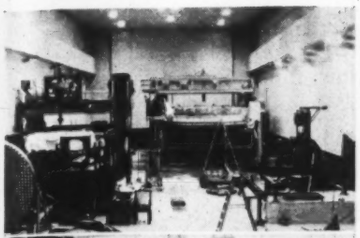
A roof duct at the Capitol building..



The central ammonia refrigerating system at St. Louis was most interesting. The compressors are horizontal, two stage type, driven by synchronous motors .....

A reversed refrigeration cycle installation was inspected at Algiers, across the river from New Orleans .....

As an amateur photographer, I took full advantage of New Orleans' Mardi Gras. The children's costumes were particularly interesting .....

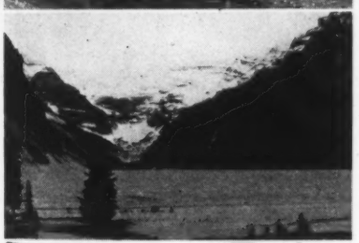


At California Institute of Technology I saw the air conditioned building where the 200 in. telescope mirror is being ground ....

Frisco, with its steep hills and cable cars, is one of the "different" cities of the states

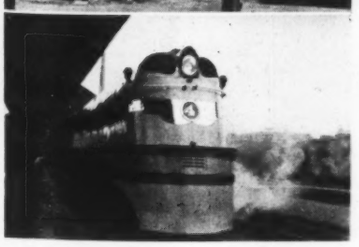
Leaving San Francisco, I wound through the redwoods .....

There can be no greater impression of concentrated power than the huge C. P. R. locomotives that haul trains over the Rockies .....



Lake Louise has been called the most beautiful in the world .....

At Winnipeg, where temperatures drop to —30 to —50, I saw the hot water district heating system. A sub-station



From Minneapolis to Chicago, I traveled on one of the new high speed streamliners. The riding was some of the most comfortable experienced anywhere .....

the relatively warm winters experienced in this district this type of job works quite satisfactorily.

The next part of the journey lay west to Houston, Texas, where numerous installations of the smaller unit type were seen. Cooling is of course an important problem in southern Texas and the demand for the small unit type of plant is heavy, as so many shops, stores and restaurants are installing air conditioning as a necessity to trade.

From Houston to Los Angeles via the Grand Canyon was a long journey but one very full of interest. Such sights as the huge, rolling plains of northern Texas, the wild desolation of Arizona and the beauty of the San Bernadino mountains created a vivid impression. Changes in climate were rapid too, as in 36 hours I passed from the midsummer heat of southern Texas to the snow-covered southern rim of the Grand Canyon.

And so to Los Angeles and, of course, Hollywood. Naturally, film studios were inspected, not forgetting the inimitable Walt Disney, and such problems as that of dealing with a casual heat load of anything up to 600 kw were discussed. The general opinion thereabouts seems to be that the ideal method of treating talkie stages has not yet been evolved, and in fact may never be; as problems of changing conditions, *i.e.* ever altering "sets" and rapidly varying casual heat loads, will always be difficult of solution, especially on the really large stages which may be anything up to 100 ft square and 80 ft high. In any case the maintenance of exact temperature and humidity conditions is not of vital importance except in sound recording stages where skin and string band instruments are in use.

Other jobs seen in this district were the Edison Building in Los Angeles and the special building at the California Institute of Technology in Pasadena where the 200 in. mirror for the new observatory near Mt. Wilson is being ground and polished. This building and its equipment was one of the most remarkable installations seen. Inside conditions can be held all the year 'round to within astonishingly small limits.

From Los Angeles I made a thousand mile automobile tour through California, visiting such places as Santa Barbara with its beautiful old Mission, and the Yosemite Valley—then snowclad. The ever changing and immense variety of the California scenery which ranged from the alpine grandeur of the Sierras to the flatness of the San Joaquin Valley with its acres of fruit farms, made a great impression upon me. Finally, San Francisco was reached via Oakland and the Oakland Bay bridge—8½ miles long. One felt rather fly-like crossing this huge structure, although in point of size the new Golden Gate Bridge is of course considerably larger. The two bridges dominate entirely the vast San Francisco Bay and it is fortunate that their designers have kept in mind their appearance as well as their strength.

Leaving San Francisco by the ferry across the bay to Sausalito in the early morning, with the sun rising, one got unforgettable views and once again was impressed by the beauty of the bridges. North through the "Redwood Empire" by train to Eureka, motor coach to Grants Pass and finally train again to Portland proved this deviation from the straight railroad route to be very well worth while, as the road follows a very much more interesting scenic route than the railroad and in addition passes right through the redwood groves.

From Portland to Seattle was another short hop via the Union Pacific Railroad. Seattle has an almost identical climate to that of England and it was interesting to observe the great similarity between the design of the installations and those of my own country. The State

of Washington has a very cheap electricity supply, and an installation seen utilizing this fact was that of two 9000 kw three phase electrode boilers for raising steam for a district steam supply system. This type of boiler is used quite extensively over the northwest section of the United States and Canada. The general arrangement is to provide each boiler with its own transformer working straight from high tension supply lines at, say, 66,000 volts. Current is supplied to the boiler at about 4400 volts from the transformer, load control being effected by lowering or raising the water level in the boiler.

A C.P.R. boat took me from Seattle to Vancouver, and here again the similarity of installations to English ones was most marked. Private house conditioning is coming in to a large extent in this district and several interesting jobs were seen, both coal and oil fired. At Vancouver I once more turned my face eastward. By C.P.R. over the Rocky Mountains to Banff and Lake Louise was of course the route chosen, and for scenic effect there can be no more wonderful views obtainable anywhere than on this line. The amazing panorama of snowclad mountains is unforgettable. Needless to say, the engineering of the railroad is in keeping with the country. Spiral tunnels, snowsheds, bridges over gorges hundreds of feet deep, and the clinging of the railroad high up on the mountain sides, all heighten the effect of scenic grandeur. In addition, there can be no greater impression of concentrated power than that given by the huge locomotives used to haul the trains.

At Winnipeg once more my attention turned to serious work and amongst other things an interesting hot water district heating system was seen. Here steam is supplied from a central station to a series of sub-stations and in these by means of heat interchangers water is heated to a temperature of about 200 F, and is then pumped through mains under the street, connections being taken to each house served. Measurement of heat used per house is performed by a meter which records water flow and temperature drop between flow and return mains simultaneously. The sub-stations are situated in the basements of various local residents' garages. Winnipeg has winter temperatures which drop to —30 to —50 F and the heating problems outweigh the cooling ones, so that air conditioning is being largely considered from the winter design point of view.

Minneapolis came next and here an instructive interview took place with Professor Rowley at the University of Minnesota, and some of the research work being carried out on heat transmission and air filtration was seen.

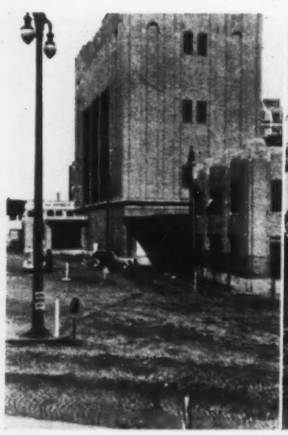
The journey from Minneapolis to Chicago was made on the "Hiawatha"—one of the new air conditioned "streamline" trains. Four hundred and twenty-one miles in seven hours means a high average speed and it speaks well for train and track that the riding was some of the most comfortable experienced anywhere. Outstanding events in Chicago during a busy week were a long and interesting interview with the secretary of the *Association of American Railroads* on air conditioning in railroad cars, a visit to the Pullman Co. works, where many cars have been air conditioned, and a visit to a chemical dehydration job in a candy factory. Then on to Detroit, where a visit to the 5000 ton air conditioning installation at the J. L. Hudson departmental store was made, as also was an inspection of the Detroit-Windsor Vehicular Tunnel. Needless to say I also spent a day at the Ford plant.

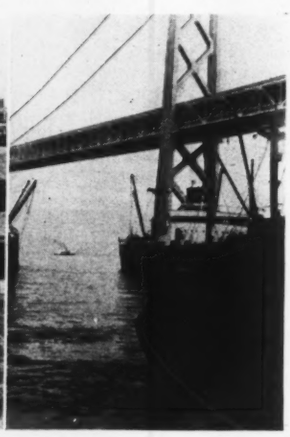
The way back to New York lay via Cleveland, Buffalo, Rochester and Schenectady and in all these towns the points of interest lay in the visits to manufacturers' works and plants. The Eastman Kodak Co. plant at Rochester was outstanding and some of the finest air conditioning anywhere was found here. Finally, New York was reached, after a journey of 11,000 miles lasting five months.

The general conclusion reached as the result of the trip was that the average standard of work seen was very high. Of course a certain amount of cheap, second rate work has been carried out, and in some cases the term "air conditioning" has fallen into disrepute as the result. This, however, is inevitable in any large industry and the formation of comfort cooling codes should help considerably to rectify matters in this respect.

Two particular features seen were of outstanding interest to me, these being the use of high speed air inlet and diffusion and the very high quality and comprehensive use of automatic controls. In connection with the latter item the modulating type of control for varying the inside-outside differential for summer cooling struck me as being very excellent, and the older idea of a fixed inside temperature irrespective of outside conditions is giving way to a much more reasonable basis of design. Also, I was struck by the average high standard of operating engineers which is certainly a good factor, as the "headaches" from a job seem to result just as often from faulty operation as from bad design.

Finally, I should like to take this opportunity of thanking all those with whom I have come in contact for their helpful cooperation; and particularly the Secretary of the *American Society of Heating and Ventilating Engineers* for his time and trouble and all the introductions with which he furnished me.





A feature of my Detroit stay was an inspection of the ventilation of the vehicular tunnel. A ventilation building .....

San Francisco was reached via Oakland and the Oakland Bay bridge, the size of which made me feel fly-like. This shot shows a ship under the bridge .....

Technical Problems in

Analyzing Fuel Consumption

By W. S. Bard*

The author recently became interested in the application of statistical methods to the comparison of steam consumption for heating buildings in order to detect tendencies toward waste . . . Supplementing an earlier description of the method, he describes this month some of the technical problems involved

ANALYSIS of data on fuel consumption for heating involves putting the data into a form permitting month-to-month and building-to-building comparisons of performance. Several methods have been developed to accomplish this reduction. One of these, the "deviation ratio," was explained and illustrated in the August H. P. & A. C.[1] The relationship between fuel consumption and degree days is described by an average or "normal" curve. An average (the quartile deviation) is then computed to describe the variation of actual consumption about this curve; and the deviation ratio (the actual deviation divided by the quartile deviation) is computed to be used for judging the efficiency of control of fuel consumption.

In general, heating consumption is determined by three groups of factors:

1. The weather.
   a. Temperature.
   b. Wind direction and velocity.
   c. Hours and intensity of sunshine.
   d. Humidity, etc.
2. The mechanical peculiarities of the building.
   a. Type of construction.
   b. Exposure.
   c. Ventilating system.
   d. Heating system.
   e. Temperature control system, etc.
3. The use of, and trend of use in, the building.

The "normal" curve, by its slope and shape; the quartile deviation, by its relative size; and the yearly treatment of trend, by raising or lowering the curve, adjust the data by removing the influences of the uncontrollable variable, the weather, and the influences of the more permanent characteristics of the building. Since the other variables of the weather are closely related to temperature, their influences are also greatly discounted. Some of the more important of the remaining variables, whose influence is still shown in the data, are:

1. The policy in the individual building concerning the elimination of wastes.
   a. The opening of windows.
   b. Leaks in the lines.
   c. Improper functioning of equipment.
2. Degree of adjustment of heating schedule to the demands of the building.
3. Improper functioning of meters.

The efficiency of the operations designed to control these variables is judged by noting the size and sign of the deviation ratio and by comparing it with other months and other buildings.

The application of this method to the steam consumption data for 68 building meters and several master meters at a group of university buildings has necessitated examination of several concepts in general use in the analysis of fuel consumption data and has unearthed several facts. The results of this investigation should prove of interest to those who work with similar data.

Examination of the "normal" curves for the group discloses three distinct types: straight line, convex upward, and concave upward (Figs. 1, 2 and 3). No general explanation of these three types of curves can yet be ventured, but the straight line seems to be most frequently found in the smaller residential type of building, the concave upward curves in buildings with predominantly indirect radiation, and the convex upward curves in buildings with predominantly direct radiation.

Theoretically, since heat loss is proportional to temperature differential, all the curves should be straight lines. Actually, the convex curve (increasing efficiency with increasing degree days) is found most frequently. This seems valid, since there is a tendency to overheat in the warmer months and to underheat in the colder months, due in part to the changing comfort demands of the occupants. As the weather becomes colder the average temperature approaches the design basis of the heating system, with greater operating efficiency to be expected. The heat lost through open windows decreases in the colder months. There is also some direct evidence that the average temperature maintained in the building decreases with increasing degree days.

Mathematical Treatment of the Curves

A general formula has been found to fit closely the three types of curves which describe fuel consumption as a function of degree days.

In some buildings steam is consumed for hot water heating, equipment uses, and line losses. Since this consumption is relatively constant and unrelated to temperature it can be designated in any particular building as the constant A (the intercept of the Y-axis when the curve is extended). Heating consumption, since it is dependent on heat loss through the exterior of the building (a direct function of the temperature differential between the inside and outside of the building) can be represented by BX, in which X is degree days and B is a constant peculiar to the building. The empirical

*Mr. Bard, a graduate of the School of Business of the University of Chicago, became interested in the problem of analyzing data on heating steam consumption as a practical application of statistical method. He is indebted to the members of the Department of Buildings and Grounds for free access to their steam consumption records.

[1] "An Accurate Method for Control of Heating Fuel Costs," by W. S. Bard, HEATING, PIPING AND AIR CONDITIONING, August, 1937, pp. 473-476.

curves show that there is also what might be called an "efficiency term," which increases in importance as the weather becomes colder. This term may be either positive or negative and can be designated as CX^2. The inclusion of a term in X^2 gives sufficient flexibility to the function so that it describes the empirical curves satisfactorily. C is constant for any particular building.

Thus the general equation for fuel consumption:

$$Y_c = A + BX + CX^2$$

in which Y_c is fuel consumed and X is degree days.

This function involves the assumption that the efficiency factor increases continuously and uniformly (that is, that there is no point of inflection and that the second derivative is a constant). This assumption is justified by inspection of the curves.

The function was fitted mathematically (least squares) to the data for five meters and indices of correlation that exceeded 0.90 were found. These high correlations indicate either that temperature is the principal consumption-determining factor in the weather, or that the other factors in the weather are so closely related to temperature that it serves as a reliable measure to which to relate fuel consumption.

The Degree Day

The degree day, which is now in general use, attempts to measure the average temperature differential between the inside and outside of buildings all over the country. A mean temperature of 65 F has been set as the average degree day base for all buildings and various investigations have been made to test the validity of this temperature as a base. Characteristic of these is the study made by the *National District Heating Association*, which indicated an average base of 66 F for 175 buildings of various types.[2] The bases found ranged from 56 to 76 F.

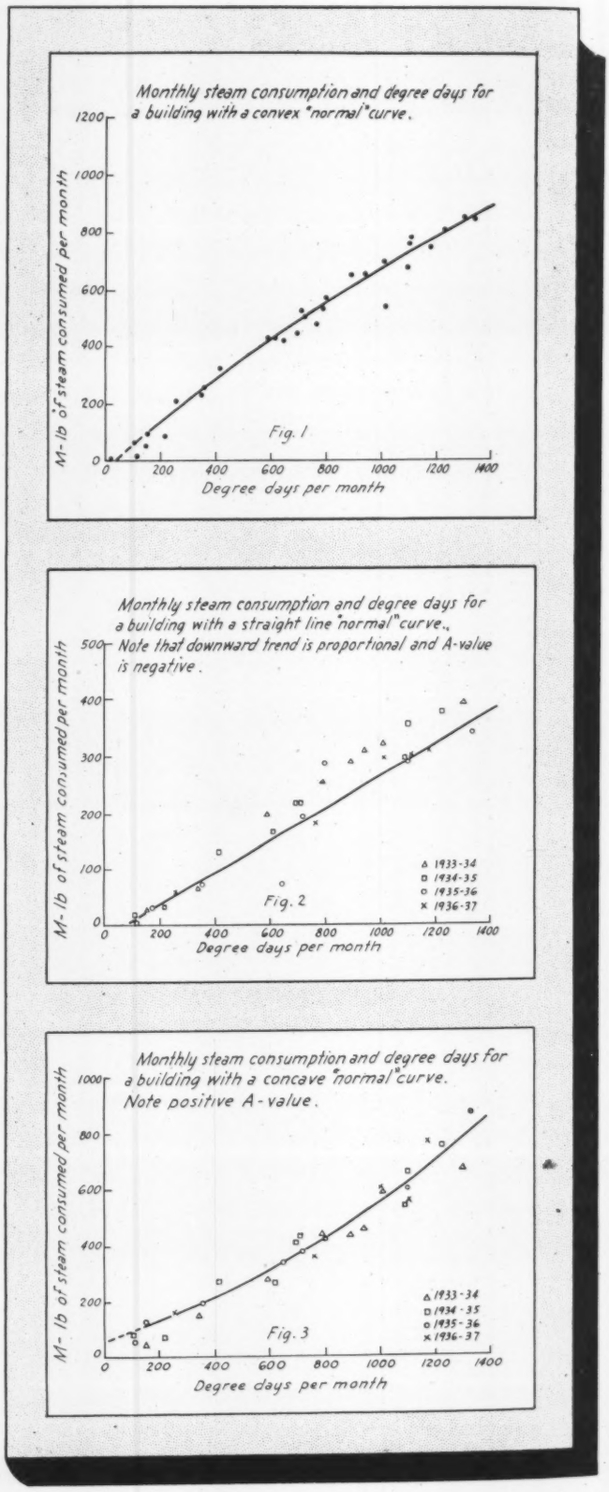
The method used to determine the base for a building is to plot average daily consumption against average temperature by months, drawing a curve to represent the relationship between the two. The data for one season are used to determine the curve. The observations in months with high average temperatures are neglected. The point where the curve crosses the temperature axis is the degree day base for the building. This method can be used only in buildings where heating fuel consumption is metered separately from other uses of fuel.

This method can be approximated by extending the "normal" curves to the base line and determining the X-value at that point. Dividing this by 30 days per month gives the amount by which the base for the building is above or below 65 F.

The "normal" curves could have been fitted against average temperature rather than degree days, but in spite of an admitted inaccuracy involved in using a standard degree day base, the degree day seemed to be the better measure. In the colder months the relative position of the observations with respect to the curve and with respect to each other are the same using either method. In the warmer months the degree day describes the relationship more closely. The standard 65 F base was used, since the slightly increased accuracy does not justify the labor involved in changing the base.

The point to this discussion of the degree day is to impress two facts:

[2] "Report of the Commercial Relations Committee of the *National District Heating Association*," 1931-1932.


Monthly steam consumption and degree days for a building with a convex "normal" curve.
M-lb of steam consumed per month
Degree days per month
Fig. 1
Monthly steam consumption and degree days for a building with a straight line "normal" curve.
Note that downward trend is proportional and A-value is negative.
M-lb of steam consumed per month
Degree days per month
Fig. 2
1933-34
1934-35
1935-36
1936-37
Monthly steam consumption and degree days for a building with a concave "normal" curve.
Note positive A-value.
M-lb of steam consumed per month
Degree days per month
Fig. 3
1933-34
1934-35
1935-36
1936-37

1. The standard 65 F degree day base is only an average for many buildings. Each building has its own average base, peculiar to it, which may be above or below 65 F.

2. The average degree day base for any particular building is only an average. The demands of the building may vary from month to month and from day to day. Hence the true base is at times above and at times below the average degree day base for the building.

Pounds Per Degree Day

The frequent use of terms such as pounds of steam per degree day, pounds of coal per degree day, gallons

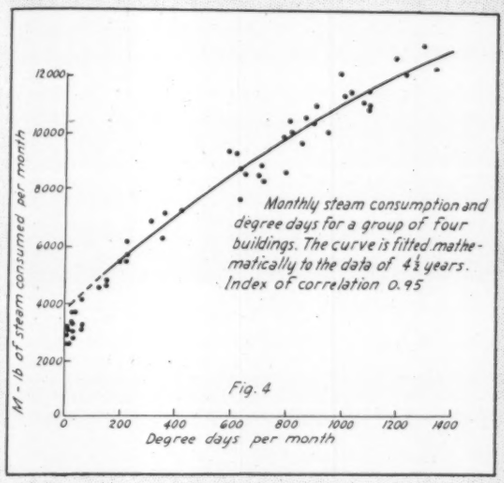

M - lb of steam consumed per month
Degree days per month
Monthly steam consumption and degree days for a group of four buildings. The curve is fitted mathematically to the data of 4½ years. Index of correlation 0.95

Fig. 4

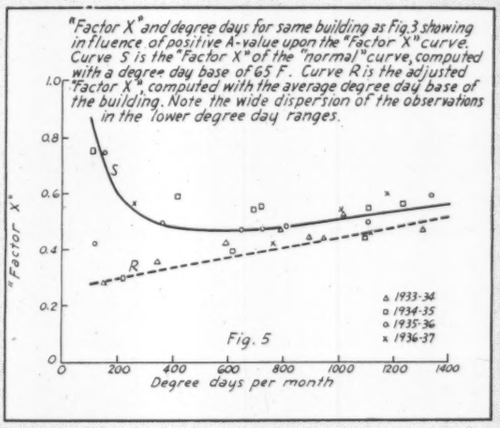

"Factor X" and degree days for same building as Fig. 3 showing influence of positive A-value upon the "Factor X" curve. Curve S is the "Factor X" of the "normal" curve, computed with a degree day base of 65 F. Curve R is the adjusted "Factor X", computed with the average degree day base of the building. Note the wide dispersion of the observations in the lower degree day ranges.
"Factor X"
Degree days per month
S
R
1933-34
1934-35
1935-36
1936-37

Fig. 5

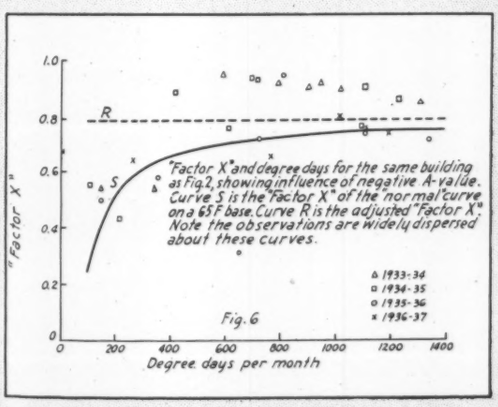

"Factor X" and degree days for the same building as Fig. 2, showing influence of negative A-value. Curve S is the "Factor X" of the "normal" curve on a 65 F base. Curve R is the adjusted "Factor X". Note the observations are widely dispersed about these curves.
"Factor X"
Degree days per month
R
S
1933-34
1934-35
1935-36
1936-37

Fig. 6

of oil per degree day, etc. attests to another method sometimes used to remove the influence of temperature from heating consumption data. In one method is used a figure called "Factor X," pounds of steam per degree day per thousand cubic feet of building space. This same figure, under different titles, is used extensively.

In "Factor X" the elimination of the influence of the variable, "size of building" is accomplished by dividing the consumption by the cubage of the building. A similar division by degree days is used to eliminate from the data the influence of variations in temperature. The resultant figure might be expected to be a constant for any particular building, on the theory that heat loss is proportionate to temperature differential, but in actual use "Factor X" varies considerably. Because of this variation, month-to-month and building-to-building comparisons of "Factor X" are not always satisfactory.

The reasons for this variation are found in what may be called "statistical" difficulties in "Factor X." We are dealing with averages, and the nature of "Factor X" is such that departures from the average are greatly magnified.

If the proper function of temperature is substituted in the "Factor X" formula:

$$\text{"Factor X"} = \frac{\dfrac{\text{Steam consumed}}{\text{Degree Days}}}{\text{1000 cu ft}} = \frac{\dfrac{A + BX + CX^2}{X}}{\text{1000 cu ft}} \text{ or } \frac{\dfrac{A}{X + (B + CX)}}{\text{1000 cu ft}}$$

The effect of the A-value, representing equipment uses, is to make the curve of "Factor X" expressed as a function of degree days the addition of an equilateral hyperbola, A/X, to a straight line, $(B + CX)$. Since A is a constant, and X a variable, "Factor X" is bound to be large in the warmer months and small in the colder months. The degree of variation caused by the A/X term depends on the relative size of the A-value, but in many buildings it is large enough to have a substantial effect (Fig. 5). Even in buildings without equipment uses there will be an A-value if the average degree day base for the building is not 65 F. The "Factor X" curve will be affected to the extent that the average base differs from the standard base (Fig. 6). The presence of an A-value in the consumption curve of any building tends to make month-to-month comparison of the "Factor X" for that building unsatisfactory unless these months have an identical number of degree days. The presence of unequal A-values in the curves of different buildings tends to make building-to-building comparisons of "Factor X" unsatisfactory.

If the average degree day base for each building were used in computing "Factor X," a material change would result in this figure (Figs. 5 and 6). In a building with an average base of 66 F the value of X in the foregoing formula would be increased 30 degree days and the A-value eliminated (if heating consumption is metered separately). The change in "Factor X" is considerable, especially in the lower degree day ranges, yet the adjusted figure is probably the more accurate.

Since in actual practice the amount of steam consumed deviates from the functionally described amount, the "Factor X" formula becomes.

$$\text{"Factor X"} = \frac{\dfrac{A + BX + CX^2 + \text{deviation}}{X}}{\text{1000 cu ft}} \text{ or } \frac{\dfrac{A}{X + (B + CX) + \dfrac{\text{deviation}}{X}}}{\text{1000 cu ft}}$$

As the size of the average deviation is not proportional to the amount of steam used, but rather tends to be constant in all ranges of degree days[1], the average effect of the term, deviation/X, is that "Factor X" deviates from its average curve more in the lower than in the higher degree day ranges (Figs. 5 and 6). This is due to the increasing size of the term as X becomes smaller. This further complicates month-to-month comparisons of "Factor X."

Still another difficulty with the factor is due to the lack of standardization of the method of measuring building cubage. Comparison of the factor for a group of buildings with another group where a different method of measuring cubage is used is misleading. To overcome

this the more standardized measurement, square feet of radiation, is sometimes substituted for cubage in the "Factor X" formula. Investigation of one group of buildings has shown that (1) the amount of steam consumed is more closely related to cubage than to radiation, and, (2) while the method of measurement of radiation is standardized, the installation of it is not. After making allowances for different exposures, varying construction, and other factors, the buildings classify into three or more distinct groups: Those that are overradiated, those that are normal, and those that are underradiated. If the influence of size of building is eliminated from consumption data by dividing by square feet of radiation, it becomes impossible to compare the three groups.

"Factor X" can be a valuable concept, but should be used with care. An adjusted factor, based on a smooth consumption curve—such as the normal curve—with the proper degree day base for each building, and with standardized cubage measurement, should prove a useful tool for research in analyzing the heating requirements of various types of buildings.

But for everyday use, the deviation ratio—which the author believes is simpler, more empirical, easier to calculate, and free from "statistical" difficulties—is more practical.

This Air Conditioner Installation Is Noisy

says a reader. Samuel R. Lewis* suggests the answer

A reader asks the following questions:

"In installing an air conditioner, suspending the unit from the ceiling in a room immediately adjoining the room being conditioned, is there a standard accepted volume of noise when the air is distributed through ducts?

"We have an installation which the customers will not accept because they claim the noise is objectionable.

"Can you make any suggestions as how to reduce noise? Are the splitters causing it and will a manufactured air splitter help? Is it good practice to install such a unit in the location this one is in? Are the grilles correct?"

In reply to these questions:

The sketch submitted (Fig. 1) indicates a small air conditioning system using a package unit fan-equipped evaporator served by a remotely located separate 5 ton condensing machine. As far as is known the plant gives adequate service except that an objectionable noise is developed by the fan. This is described as "wind noise."

The twin fan unit, which delivers about 2000 cfm, is placed directly behind one of the three air supply openings; any air pressure sufficient to force delivery through the two branch ducts would result in too great a volume through the central outlet unless the latter were throttled.

There are installed two diverging solid splitters intended to accomplish this result, each arranged so that about three-quarters of the outlet area from each fan in the unit is diverted toward the respective longer duct. The net result of this construction is to form a megaphone-like chamber behind the center discharge grille, expanding toward the room, and intensifying the noise. There is nothing new in this phenomenon, though perhaps there has not been sufficient publicity given to it.

The instructions published by all manufacturers of this sort of equipment should carry a warning against the use of ducts of unequal length without some provision for equalization and noise absorption, and a warning against outlet ducts flaring toward the discharge grilles.

In the case cited there are two reasonably hopeful solutions:

1. Close the central discharge opening entirely and remove the splitters, forcing all of the air to the two branch openings. Line the interior of these ducts for, say, 6 ft toward the fan with noise absorbing material.

2. Move the unit a distance of, say, 6 ft farther away from the center opening, building between the unit and the duct a metal chamber as wide and as deep as the unit, Fig. 2. This is to have a manhole door. The three ducts can take off from this chamber at comparatively low velocity under static pressure. The interior of the chamber is to be lined on all sides and top and bottom with noise absorbing material. If the noise still is objectionable, a honeycomb type of noise absorber can be added within the chamber opposite the fan discharge openings. Such a device might have open end cells about 4 in. x 4 in. and 24 in. long, interposing very little added resistance to air flow.

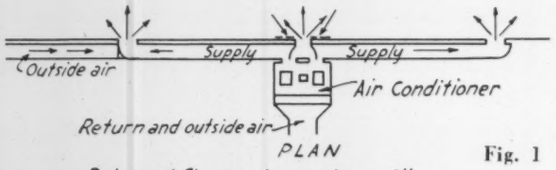


Fig. 1

Return at floor under center grille.
Splitters at center grille soldered.
Cfm through supply ducts 900 cfm.
" " return " 800 "
" " outside air " 800 "

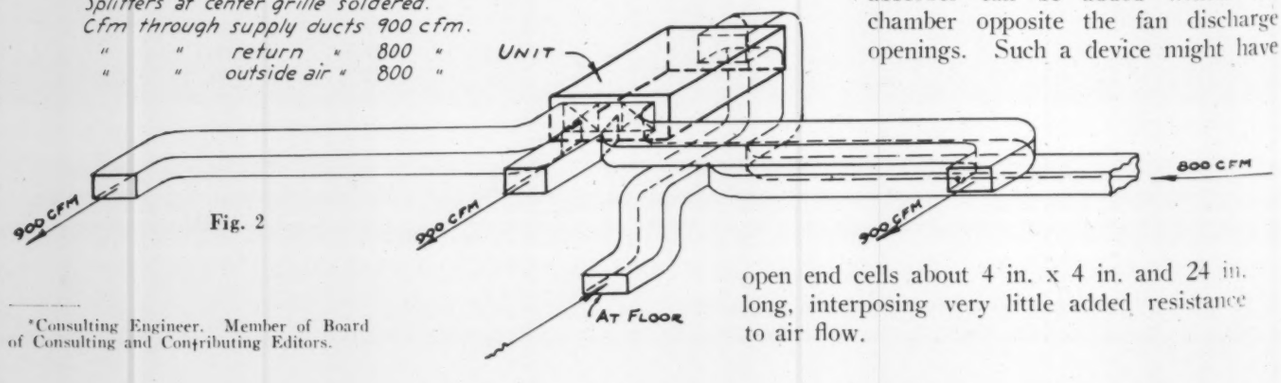


Fig. 2

*Consulting Engineer. Member of Board of Consulting and Contributing Editors.

Formulas for

Stresses in Bolted Flanged Connections

By E. O. Waters*, D. B. Wesstrom**, D. B. Rossheim†, and F. S. G. Williams††

NONE of the formulas for axial stress at the outer surface of hub or shell take into account any stress concentration that may exist because of sudden changes in section, specifically sharp corners at the junction of hub and ring. For example, when the hub is of uniform thickness ($a = 0$; $g_1/g_0 = 1$), the flange section is indicated by *knl* in Fig. 9 and Equation 8 will give the axial stress at point *n*, but without including the additional stress at that point due to the sudden change in section, which corresponds to the "notch effect" found in similar mechanical structures. If the flange is of cast metal, a substantial fillet at this point is practically essential to the proper grain formation to produce a sound casting, but apart from this, mechanical design dictates the need of a fillet *fm*. For this reason the *A.S.M.E.* code requires a fillet at the junction of ring and hub (or shell) of at least one-quarter of the thickness of the hub (or shell) and in no case less than $\frac{3}{16}$ in. The additional cross-section *fmn*, shown shaded in Fig. 9, is not regarded as strengthening the flange, except to neutralize the "notch effect," and is not considered when applying Equation 8 and related formulas.

It is obvious, however, that a larger fillet, such as *abc* in the figure, is more than required to counteract stress concentration and serves to strengthen the flange considerably. In effect it is a hub of circular section, but in lieu of developing a method of stress analysis for such a section, which would be extremely complicated, it is suggested that the fillet may be considered as equivalent to a tapered hub with an angle of 45 degrees to the axis of the flange, and tangent to the fillet, as indicated by the section *kdel*. Denoting the radius of the fillet by *R*, this results in values of h and g_1 as shown in the figure. The excess cross-section denoted by the shaded areas *abd* and *bce* is not assumed to strengthen the flange, these areas being regarded as neutralizing the "notch effect" at points *d* and *e* respectively.

In all practical cases where fillets are made large enough to reinforce the flange section, it is estimated that the maximum axial stress will always occur at the small end of the "equivalent hub" *kdel*. This proves to be the case in the investigation presented later in this paper, the maximum stress being found in all cases to be located at point *d*, a distance of 0.585 *R* from the ring. This is almost identical with the distance 0.6 *R* estimated by Jasper, Gregersen and Zoellner (7)§ from their tests on plaster of paris models, from which they derive a modification of Timoshenko's formula with which a comparison also is made below. The agreement obtained is quite remarkable when considering that in the two methods, one analytical and the other experimental, the approach is totally different.

Large circular fillets, however, are not believed to constitute economical design. Referring to Fig. 9, if the minimum required fillet is used at *d* and *e*, the bolt circle can be moved inward, thus reducing the moment on the flange, while retaining a hub *de* of equivalent strength. Conversely, if for other reasons the bolt circle cannot be changed, a stronger hub can be had by providing the minimum fillet at *c* and using a hub section *a i c*, or an even longer one *k j c*.

Previous to this paper, no formulas based on elastic theory were available for tapered hub flanges with integral shells of uniform thickness. Formulas had been developed, however, for some special cases.

For example, for loose ring flanges without hubs Equations 8 and 9 are non-existent and the second term in the right hand side of Equation 10 vanishes. The resulting equation is found to be identical with the formulas presented by Waters and Taylor in 1927 (1), Equation 1 on page 534. It is also identical with Holmberg and Axelson's formula for the condition of loading assumed in this paper.

Waters and Taylor in their 1927 paper also presented formulas for flanges with hubs of uniform thickness ($a = 0$; $g_1/g_0 = 1$). These are similar to but not identical with Equations 8, 9, and 10 of this paper when the

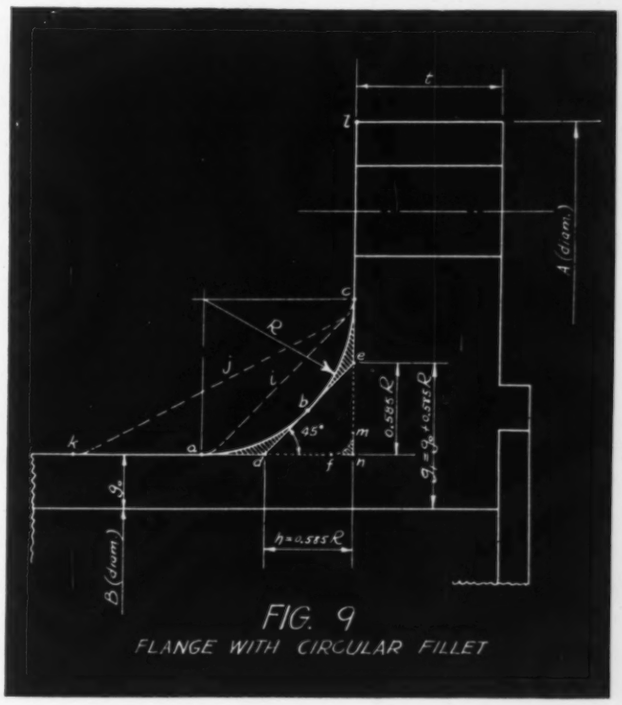


FIG. 9
FLANGE WITH CIRCULAR FILLET

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††District Manager, Taylor Forge and Pipe Works.

Parts 1, 2, and 3 comprised a paper contributed by the power division for presentation at the semi-annual meeting of the *American Society of Mechanical Engineers*, Detroit, Mich., May, 1937, and published by permission.
Part 1. Part 1 was published in June, pp. 363-366, Part 2 in July, pp. 431-434, and Part 3 in August, pp. 485-487.
§See bibliography.

latter are reduced to this special case. Minor but appreciable differences in the results are due to a more complete analysis in deriving Equations 8, 9, and 10.

Other formulas covering this special case were given by Timoshenko (4), and by Holmberg and Axelson (6), and in a series of articles recently published in HEATING, PIPING AND AIR CONDITIONING (7) Jasper, Gregersen and Zoellner compared these formulas with the results of tests made by them on plaster of paris models of hubbed flanges. They also made additional tests to investigate the effect of circular fillets at the junction of hub and ring and then developed a modification of Timoshenko's formula to apply to such designs. The test results they obtained from the plaster of paris models are a distinctly valuable contribution to the subject of flange design and indicate substantial agreement with theory.

Two series of tests have been selected for comparison with Equations 8, 9 and 10, the first represented by Figs. 10 and 12 in the above mentioned articles, and the second by Figs. 13 and 14 therein.

In the first series, the ring thickness t is variable, all other dimensions remaining the same throughout, as follows:

$A = 16.35$ in.; $B = 10.9$ in.; $K = 1.50$; $g_1 = g_o = 0.8$ in; and $h' = 4$ in. = total axial length of flange minus the ring thickness. The value of $h'/\sqrt{Bg_o}$ is 1.35 so that these may be regarded as integral flanges. In applying Equations 8, 9 and 10, the following constants are used:

$T=1.72$; $U=5.46$; $Y=5.00$; $Z=2.60$; $F=0.909$; $V=0.550$; and $f=1$.

The foregoing is for a zero fillet radius R at the junction of ring and hub. For $R = 1$, which was the largest size of fillet tested, using the method described above, h becomes 0.585 R or 0.585 in. and $g_1 = 1.385$; $g_1/g_o = 1.73$; and $h/\sqrt{Bg_o} = 0.198$. The constants are therefore changed as follows:

$F = 0.884$; $V = 0.394$; and $f = 1.96$, indicating that the maximum axial stress is at the smaller end of the "equivalent hub." The quantities T, U, Y and Z are the same as before.

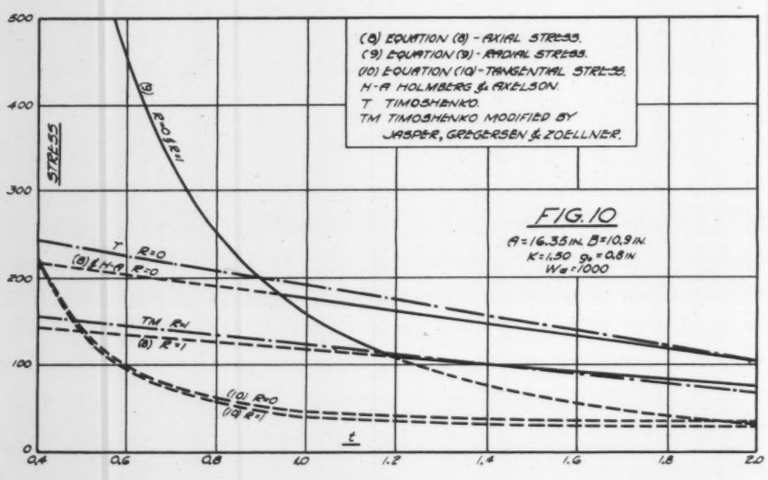
These results are plotted in Fig. 10, in which the ordinates are the stress produced by a unit external moment Wa on the flange, assumed for convenience to be 1,000 in.-lb. Equations 8, 9 and 10 are drawn with dashed lines except the part of each equation that determines the maximum stress in the flange as a whole, and this is drawn solid.

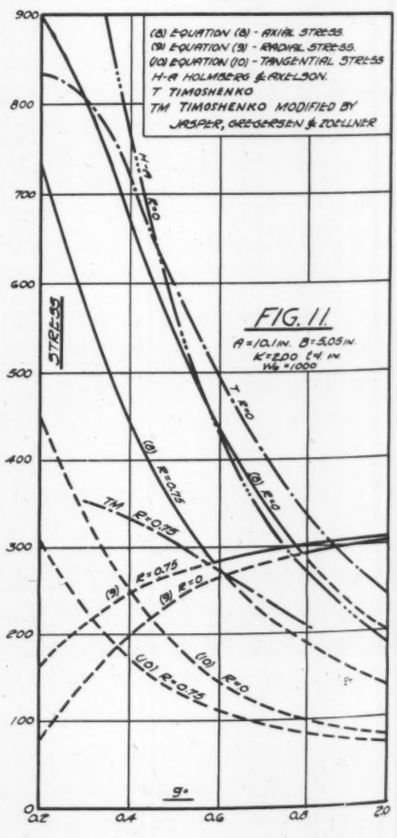
For $R = 0$, the Holmberg-Axelson formula coincides almost exactly with Equation 8, and therefore these are drawn as one line. There is good agreement also with Timoshenko's formula. Equation 10 is not critical, but Equation 9 for the radial stress becomes critical for the smaller flange thickness, as would be expected.

For $R = 1$ there is a substantial reduction in the axial hub stress in Equation 8, which agrees very well with the modified Timoshenko formula. However, there is very little reduction in the tangential stress, Equation 10, and practically none in the radial stress, Equation 9, so that the provision of a fillet cannot be said to strengthen the flange when the ring is thin.

In the second series of tests considered, g_o and g_1 are variable and all other dimensions constant, as follows: $A = 10.1$ in.; $B = 5.05$ in.; $K = 2.00$; $t = 1$ in.; and the distance h' from the ring to the end of the shell is 4.5 in. The value of $h'/\sqrt{Bg_o}$ is always great enough to warrant considering these as integral flanges. For Equations 8, 9 and 10, and with $R = 0$, the constants are: $T = 1.514$; $U = 3.25$; $Y = 2.95$; $Z = 1.67$; $F = 0.909$; $V = 0.550$; and $f = 1$.

For $R = 0.75$ in. the largest size of fillet tested in this series, h becomes 0.439 in. and g_o, g_1, g_1/g_o, $h/\sqrt{Bg_o}$, F, V and f are all variable. The value of f is greater than unity in all cases, indicating that the maximum axial stress is at the smaller end of the "equivalent hub."





Progress in pressure vessel design and fabrication during the past two decades has brought about a marked change in the form of bolted connections. Whereas flanges were formerly provided with hubs of approximately uniform thickness, screwed to or slipped over the shell or pipe, it is now customary to use tapered hubs, and an integral structure is obtained by butt welding the end of the hub to the shell. This has invalidated former methods of design for flanged connections. . . . . This paper outlines a revised analysis based on the ring, tapered hub, and shell being considered as three elastically coupled units loaded by a bolting moment, a hydrostatic pressure, or a combination of the two. Design formulas and charts are given for the computation of stresses that are likely to be critical, and their application is illustrated by three problems which are taken from current commercial practice

The constants T, U, Y and Z are the same as before.

The results are plotted in Fig. 11 on the same basis as used in Fig. 10. For $R = 0$, there is good agreement between Equation 8 and the Holmberg-Axelson and Timoshenko formulas. Equation 10 for tangential stress is not critical, but Equation 9 for radial stress becomes critical at large values of shell thickness g_o.

For $R = 0.75$, there is a slight increase in radial stress, and this stress remains critical at lower values of g_o because there is a substantial reduction in axial stress in Equation 8. There is also a reduction in tangential stress in the ring, Equation 10. The agreement between Equation 8 and the modified Timoshenko formula disappears at low values of shell thickness, and this is probably due to the fact that the Timoshenko formula as modified by Jasper, Gregersen and Zoellner does not take into account the excessive stiffening effect of the relatively large fillets in this region, which requires the bending action to take place almost completely in the shell.

It must not be inferred from Figs. 10 and 11 that the tangential stress is never critical, as this feature has not been explored sufficiently to warrant such a general conclusion.

In 1927 Timoshenko (4) also presented formulas for flanges with shorter hubs of uniform thickness, equivalent to the loose hubbed flanges in this paper (solved by Fig. 5). A comparison with Equation 8, using the example discussed by Timoshenko, shows substantial agreement.

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[*Concluded*]

Refrigeration Piping—

[*Concluded from page* 536]

usually very short. If the suction line is a riser, the velocity of the suction gas should be at least 1000 fpm to be sure of good oil recovery. If the suction line is too large, the resulting low velocity may cause trouble in oil recovery and the cost of the line will be higher than it needs to be. If the suction line is too small, the resulting gas velocity will be extremely high with a high pressure drop and a consequent falling off in compressor capacity. It is, therefore, good engineering in designing "Freon" pipe and valve sizes to keep within established limits and bounds in order to maintain piping cost at a sound and economical low point, and yet avoid operating difficulties.

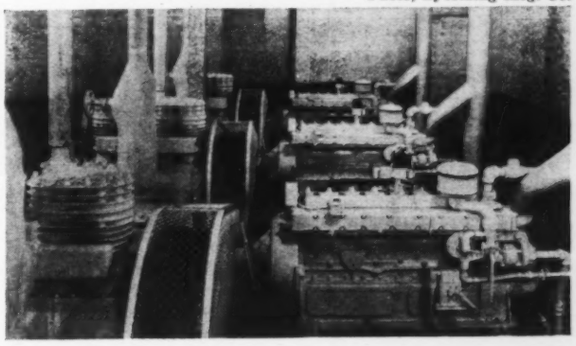
Natural Gas Engines Drive Compressors

A FEATURE of the gas industry exhibit in the varied industries building at the Pan-American Exposition, Dallas, Texas, is the battery of natural gas engines that supply power for the building and drive the air conditioning compressors.

The electric plant consists of three 60 kw engine driven generators, which develop a total of 180,000 watts. Three compressor units, which air condition the building, are each driven by a 50 hp gas engine connected by a V belt drive. Engine speed is automatically controlled to maintain a given temperature in the building.

Three 50 hp gas engines driving air conditioning compressors

Photo, Lycoming Mfg. Co.



Condensing Water for Air Conditioning City Hall Is Put Back in Supply Main

By **Joe Stocker**

WITH the problem of supplying and getting rid of condensing water for the refrigeration equipment of air conditioning plants becoming a factor to which more and more attention is given, the scheme adopted for Oklahoma City's new city hall is of interest. Morrison B. Cunningham, city water superintendent, and Frank Herrmann, water engineer, decided to tap a 20 in. main northwest of the building, pipe the water through the condensers and back into the main as it curves around the building in a quarter-circle southeastward.

A 6 in. extension was laid westward 110 ft from the city hall to join the 20 in. main running north and south. Two 700 gpm circulating pumps pump the water into two condensers having 2400 sq ft of heat transfer surface. Then the water is pumped out via an 8 in. extension to rejoin the main 300 ft southeast of the building. The average pressure on the 20 in. main of 75 lb per sq in. is increased 20 lb through the pumps.

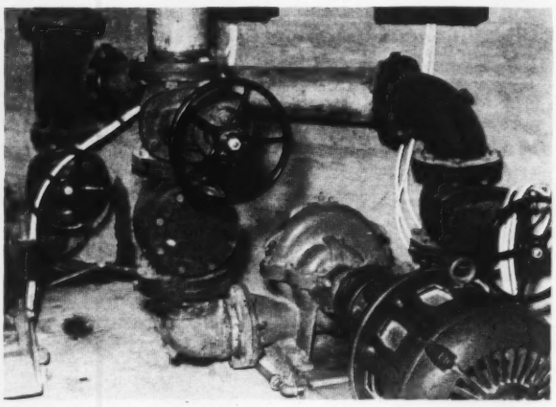
Pending the scheduled installation of recording thermometers and a flow meter for hourly and continuous readings on 24 hr charts, municipal engineers are checking water temperatures with indicating thermometers. In the first week of operation, the water temperature increased 4 F or slightly less through the condensers, with one compressor and one pump operating. With two compressors and one pump, the temperature increases 8 F, and with both compressors and both pumps, 4 F. Except in extremely hot weather, only one compressor and one pump are necessary.

The average condenser water temperature increase is from 79 F at the inflow to 82½ F at the outflow.

Aware of the commercial possibilities of the method, municipal engineers are watching closely for contamination. Hourly water samples are taken from each condenser, placed in sterile bottles and analyzed by a water department chemist for ammonia pollution or any other extraneous matter. So far there have been no traces of contamination.

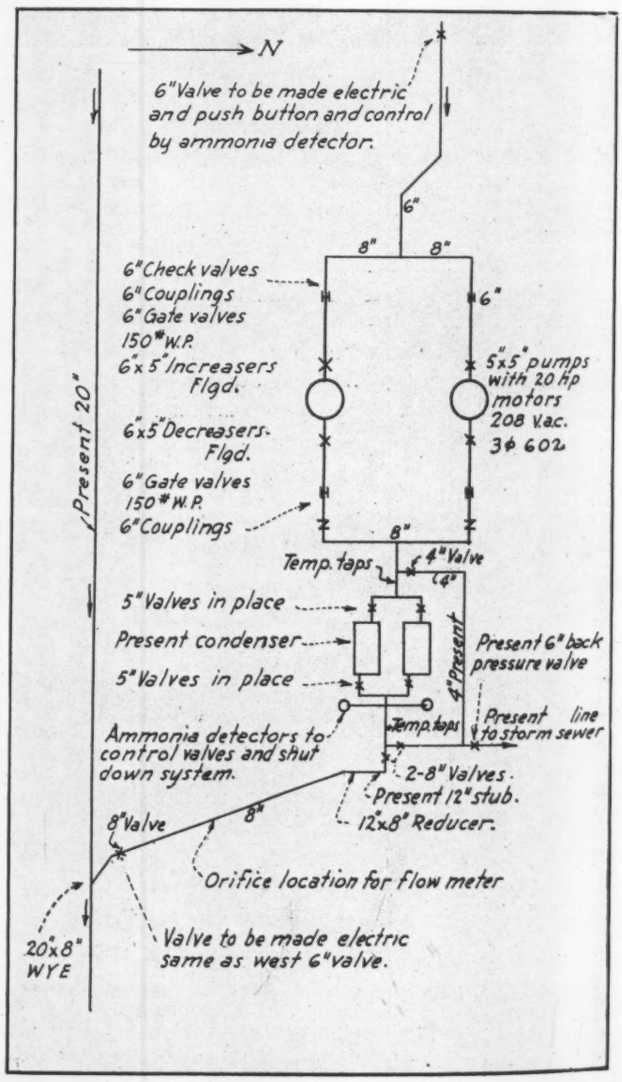
An attempt was made to obtain an instrument to detect moisture in the ammonia line and shut off the compressors automatically at the first evidence. A leak in the condenser tubes would cause moisture to be pulled into the ammonia lines. Hand tests will continue, meanwhile, until apparatus can be adapted to this method of control.

Special permission to tap municipal mains was obtained from the state health department, with the understanding that the connection is to remain under the supervision of the water department. Should private installations be permitted, city officials propose that condensers, valves and pumps be sealed by the water department and extra inspectors be employed by the city to safeguard against pollution.



Two pumps force the water through the condensers

Schematic drawing of the condensing water piping



At Oklahoma City's new city hall, condensing water for the air conditioning plant is taken from the city main, passed through the condensers, and returned to the main. . . Under the supervision of the city water department, tests of possible contamination and temperature readings are being taken, and the idea of permitting private installations is being considered

When breakdowns occur, inspectors only should be empowered to break the seals and supervise repairs to avoid pollution. After the system is opened for repairs and before operation is resumed, the system should be thoroughly sterilized with a portable chlorinator or other suitable means of introducing chlorine.

Two methods of charging for private use of the water are suggested. There might be a standard charge for each 1,000 Btu's, or a monthly service charge for the amount of water used. Since there is relatively no water consumption, however, the former schedule is deemed the more feasible. Revenue from such sources would meet the cost to the city of employing special inspectors and making periodic chemical tests.

The Plant

R. W. Schulte, Oklahoma City consulting engineer, designed the refrigeration plant. The 125 ton compressors are single acting, two cylinder, enclosed type, with 10½ in. bore by 10½ in. stroke, running at 250 rpm. Suction pressure is 60 lb and discharge pressure 160 lb.

The ammonia receiving tank is 2½ by 14 ft. The shell and tube evaporator, which cools the water used in the heat exchangers in the four fan units, has 2200 sq ft of heat transfer surface. This water enters the evaporator at 46 F and emerges at 41 F. Each of the four fan units has a capacity of 20,000 cfm.

The installation cost of piping water into the condensers from the mains was about $3500. Private installations on the same pattern would cost a little more because of the necessity for more safeguards in the absence of constant city supervision.

Related Reading

This method of supplying and disposing of condensing water was proposed in some detail in an article by E. Vernon Hill published in Heating, Piping and Air Conditioning, April, 1937, pp. 233-236. A number of other articles of related interest have appeared in recent issues of H. P. & A. C. Among them have been "Effect of City Water Supply and Sewer Systems on Air Conditioning Reported," August, 1937, pp. 480-483; "Use of Sewage Water as Condensing Medium for Air Conditioning Plants Has Possibilities," by E. P. Heckel, December, 1936, p. 653; "Effect of Air Conditioning Demand on City Water Supply and Sewer Systems," by Loran D. Gayton, November, 1936, pp. 609-612; and "Air Conditioning Makes Severe Demands on Water Supply and Disposal Systems," September, 1936, pp. 491-494. The last mentioned article included data by L. L. Lewis on water consumption, and a proposal to use raw water for condensing purposes, by Walter L. Fleisher.

Looking Ahead in Railroad Air Conditioning

By C. C. Elmes*

AS of October 1, 1936, there were 28,738 passenger cars owned by the Class I railroads of the United States. The total number of miles of road operated was 233,428. Many of these cars are of wooden construction, unsuitable for air conditioning due to age and improper insulation. An undetermined number are in suburban service; because of the opening of doors at frequent stops, the installation of air conditioning on such equipment is questionable from an economic viewpoint.

As of January 1, 1937, there were 4231 railroad-owned and 4124 Pullman-owned cars air conditioned, a total of 8355, representing an expenditure of more than $50,000,000 for equipment and necessary terminal facilities. In 1932 there were only 50 cars air conditioned. It is estimated that some 1000 cars will be air conditioned during 1937, representing an expenditure of from $4,000 to $8,000 per car, depending upon the type of air conditioning system used.

The Division of Equipment Research, *Association of American Railroads*, has concluded a comprehensive study of the air-conditioning systems and appurtenances now in use.[1] The work done in 1936 not only developed information on the performance of air conditioning equipment under controlled laboratory conditions, but also on the performance in actual road service. With information gathered from 594 separate runs over 31 railroads, and from laboratory tests of 15 air conditioning systems and 14 air conditioned cars, it is possible to base certain definite conclusions as to what should be done in the future with respect to the air conditioning of railroad passenger cars.

One should be mindful that there is always the possibility of some new development which may upset the trend of past thinking and practice. However, refrigeration is nothing new, nor is it an art which has not already gone through years of development and improvement. It is true that the application of its fundamental principles to a railroad car calls for changes in design, but the same basic principles hold. There is nothing complicated about a railroad air conditioning unit, merely the usual combination of heat exchange devices, and there is no reason why these devices cannot be manufactured upon a production scale. The main future development, therefore, should be volume production, resulting in lowered purchase price. Some manufacturers have this in mind and are developing their equipment for either private

*Association of American Railroads.

[1] See Heating, Piping and Air Conditioning, June, 1937, p. 118 (back section).

homes or railroad service. Considerably reduced prices are being quoted upon orders of several units, which shows that even now a moderate volume of business will reduce costs.

As pointed out by the Division of Equipment Research in a cost study made of air conditioning as applied to railroad cars, the major item is the fixed charges, based upon the purchase price and the installation cost.

The cost of operation of an air conditioning system, whether installed upon a railroad car or not, for a given capacity, cannot be reduced to any great extent. A reduction in the price of ice would naturally lower the operation cost of the ice activated system, but those systems using mechanical, electrical, or steam power cannot expect any great reduction. It requires a certain theoretical horsepower to produce a ton of refrigeration and for small installations, any economy in power that can be effected will be relatively unimportant from a cost standpoint.

The average mechanical compression air conditioning system on a railroad car requires up to 25 hp from the locomotive for full capacity operation. A train equipped with 10 air conditioned cars has an additional pull on the locomotive due to air conditioning of about 5-10 per cent of its available power. The future trend may be to lessen this added load on the locomotive by the use of self-contained units which are entirely independent of the locomotive for power. This would permit the operation of longer trains of air conditioned cars than is possible when dependent upon the locomotive alone for power.

The cost of maintenance is an item that should be reduced. In most cases, with railroad air conditioning units, the cost of maintenance is more than the cost of operation.

The mechanical compression, ice activated, and steam jet systems, all founded on well grounded principles, will no doubt continue to be favored, as will the ice activated type in those regions where a plentiful supply of ice may be had at reasonable prices. The ice system has the additional advantage of lowest first cost and lowest maintenance cost of all the systems now in use. Systems employing dry ice, except in rare instances, will probably be prohibitive because of the high operation cost. Another means for air conditioning, the absorption system, is limited in its application because of the large amount of space required for a unit of sufficient capacity to cool a railroad car.

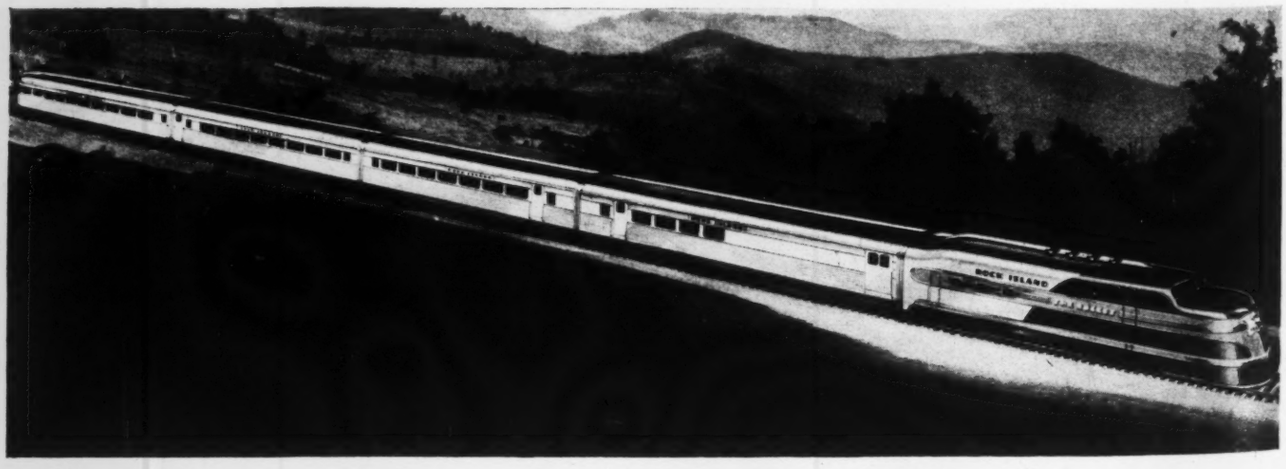
Improvements in the mechanical compression system should include reduced discharge pressures at high outside temperatures and more adequate condenser capacity. Evaporative cooling of the condensers and an efficient means of subcooling the liquid refrigerant are rapidly coming into more favor.

The future air conditioned railroad cars will have more positive control of temperature and humidity in both winter and summer. The controls should be entirely automatic, and the five essential parts of air conditioning—heating, cooling, ventilation and cleaning, humidification, and dehumidification—should be interlocked so that whatever condition is called for will be furnished independent of any human adjustment.

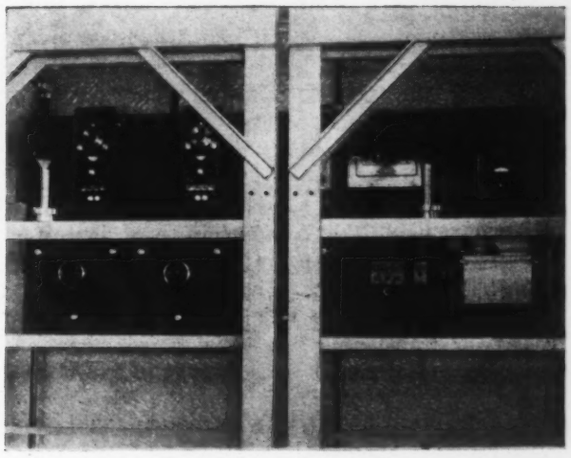
Improvements may be made in the air cleaning devices now used, as the replacement and cleaning expense of filters is a considerable item. Electrical filtration of air is a distinct possibility. The Division of Equipment Research is conducting a thorough research on filters, using the laboratory facilities of the University of Minnesota.

Efforts should be made toward the reduction of the common complaints found among passengers in air conditioned cars. A survey of passenger reactions in 1936 made by the Division of Equipment Research shows that 435 out of a possible 2822 unfavorable comments indicated that there was too much heat from sunshine through the windows; 240 comments, that there was a stuffy condition within the car; 197 comments, that the car was drafty; and 140 comments, that there was an objectionable odor. The air conditioned passenger car of tomorrow will evoke none of these common complaints, and their absence is an indication as to whether the car is truly "air conditioned."

Artist's drawing of one of the Rock Island Lines' new streamlined, diesel powered "Rockets". The stainless steel cars are fully air conditioned



Taking a measurement at creep furnace



Controls and potentiometer recorder

Creep Testing of Metals

By J. J. Curran*

IN all high temperature, high pressure applications, the "creep" or permanent deformation of the structural metals in service is extremely important. To meet the higher temperatures and stresses involved it has been necessary to examine closely into the relative merits of the various commercially available materials, and to develop modifications of the more common materials to attain the maximum of resistance to this deformation.

The creep test used to evaluate the quality of various materials for these increased combinations of temperature and stress consists in heating a specimen of the material to the proposed service temperature, applying the contemplated stress, and observing the deformation which ensues. In theory, this is simple enough, but in practice complications set in.

The creep requirements which must be met in the valve and fitting industry are varied. In some applications obsolescence occurs in a relatively short period, due to causes other than deformation in service. In other applications, longer useful life requires an extremely low rate of creep, not in excess of 1 per cent per 100,000 hr.

A creep rate of this magnitude amounts to a change in dimension of one ten millionth inch per inch per hour, or twenty-four ten millionths inch per inch per day. If it is remembered that the linear expansion in an ordinary steel is of the order of seventy-four ten millionths inch per inch per degree Fahrenheit, it is evident that a fluctuation of only 1 F in the temperature of the specimen from day to day will produce a change in length approximately three times as great as that produced by the actual creep in the specimen in 24 hr at the 1 per cent per 100,000 hr rate. These figures will serve to indicate the necessity for extreme accuracy in control of temperature in the specimen as well as in measurement of the deformation produced.

*Research Metallurgist. Walworth Co. Reproduced by permission from *Walworth Today*.

Since it is not feasible to run tests for periods as long as 100,000 hr, we must resort to shorter tests. The question then arises as to the minimum time in which a sufficiently accurate value may be obtained for extrapolation to the longer period. Important limiting factors are the accuracy of temperature control and accuracy of length measurement. The shorter the test period, the more accurate must be both of these factors. In extended tests, the effects of slight inaccuracies in these directions are minimized; the longer the actual duration of the test, the less important becomes the error.

Modern demands for higher efficiency in many industries have led to the development of plants and processes in which temperatures and pressures have been increased far beyond those commonly used only a few years ago. In high temperature, high pressure applications, the "creep" or permanent deformation of the structural metals in service is extremely important . . . Laboratory technique in studying creep is described briefly here

Since the inception of creep testing about 15 years ago, there have been constant improvements in testing technique, with consequent improvement in the accuracy of the values obtained. Improvement in the technique has been so rapid and so marked that methods and values considered satisfactory only a few years ago are now condemned as entirely inadequate for extrapolation to long periods of service.

The creep testing equipment and technique of the general metallurgical research department of the author's company have been designed and refined to secure the highest attainable accuracy of results, and to minimize the errors found in creep testing in the past, to the end that the values obtained will permit of extrapolation to any reasonable service life. It is believed that engineers

who use materials where creep is a factor will find this brief description of research into the subject of interest.

The creep test machine is somewhat novel in design, employing the principle of direct loading to avoid the errors inherent in multiplying levers due to friction, temperature variations within the laboratory, and eccentricity of loading.

The loading system consists of two concentric cradles designed to hold 6,000 lb of weights, sufficient to produce a stress of 30,000 lb per sq in. in the standard 0.505 in. diameter test specimen. The live load is carried in one cradle, and the excess weights in the other, with provision for transferring the weights from one cradle to the other to increase or decrease the stress on the specimen.

The test specimen used is standard. The telescoping platinum strip method is used to provide reference marks for measuring elongation or creep during the test. The diameter of the specimen is 0.505 in., with an area of 0.200 sq in. Holes are drilled axially in the grip ends of the specimen to accommodate thermocouples to control and record the temperature of the specimen throughout the test.

At the left of the diagram is shown the calibration specimen, used for exploring the temperature distribution within a furnace. In addition to the axial holes in the grip ends, holes are drilled radially in the gage length of the specimen to determine the distribution over the section which carries the load.

Each furnace, before starting a test, is placed on the calibration stand with a calibration specimen in place and heated for a sufficient period to permit adjustment of the furnace to give the desired uniformity of temperature distribution, before being placed on a test stand for actual test.

Several types of electric furnaces are used for heating. One of these is the more commonly used "gap-wound" type, consisting of a refractory tube surrounded by insulating material and wound with a wire resistor so spaced as to apply heat at the ends of the furnace, where heat losses by conduction are greatest. A silver inner tube is provided to effect more uniform distribution of temperature of the gage length of the specimen.

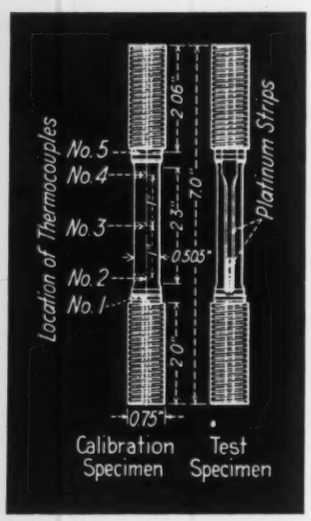
Another type is the more recent "tapped and shunted" type in which the resistance wire is wound uniformly over the length of the furnace, but is provided with taps and shunts by means of which the temperature over the length of the furnace may be compensated and adjusted.

These creep tests are run a minimum of 2,000 hr at temperature under load. During this period the temperature throughout the specimen is maintained uniform and unvarying within close limits.

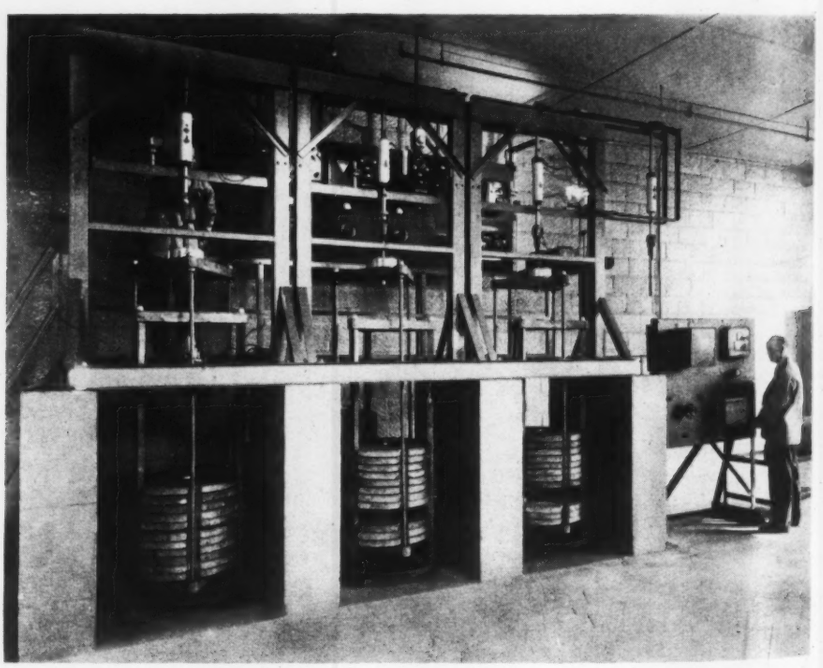
To achieve this, each unit is provided with an automatic temperature control in the form of a potentiometer controller, which cuts in and out sufficient current to maintain constant temperature within these limits. The potentiometer recorder also furnishes a continuous record of the temperature of the specimen throughout the test, serving as a check on variations in line voltage and accidental shut-downs during the night when no observers are on duty.

Measurements of stretch or deformation are made by means of an accurately calibrated microscope, sighted through a quartz window in the wall of the furnace on to hair-line scratches on the telescoping platinum strips attached to the test pieces. These measurements are made twice daily throughout the duration of the test.

These daily readings are plotted in terms of extension per inch of length, and from the curves the rate of creep is determined. Usually the slope of the curve in the second thousand hours of a 2,000 hr test period is taken as the creep rate.



Test specimens



General view of creep test machine

Practical Standards for

Air Conditioning Calculations

Outside Air Requirements and Refrigeration Load

By **W. S. Bodinus***

THE quantity of outside air required for ventilation depends upon such factors as the volume of the space, kind of occupancy, number of people, and length of occupancy. The nature of the use of the building also has its effect upon the correct amount of outside air since department stores, five and ten cent stores, and many other establishments leave the doors open and sufficient outside air must be supplied to offset infiltration as much as possible. Ordinances have in many cases established minimums for ventilation and of course these must be adhered to. Generally the amount of outside air required by ordinance is determined on one of the following bases:

(1) *Number of changes of volume of room per hour.*

If the requirements are six changes per hour, this would amount to 100 cfm in a 10 × 10 × 10 ft high room.

10 × 10 × 10 = 1000 cu ft.

1000 cu ft × 6 = 6000 cu ft per hr.

6000 ÷ 60 minutes = 100 cfm of outside air.

(2) *Quantity of outside air per square foot of floor area.*

If the requirements are 1.2 cfm per sq ft of floor area, this would amount to 120 cfm in a 10 × 10 × 10 ft high room.

10 × 10 = 100 sq ft floor area.

100 × 1.2 = 120 cfm outside air.

(3) *On the basis of cubic feet per minute per person.*

If the requirements are 25 cfm of air per person and if three persons occupied the room, this would amount to 3 × 25 = 75 cfm outside air.

One problem that is ever present is the certainty of delivering the outside air to the points needing ventilation. This is specially true where ventilation standards are based on the total people occupying the space and where the number of people is small. Also, infiltration and volume of building will have a certain effect on the amount of outside air necessary. The general minimum standard of outside air is as follows:

Schoolrooms	25	cfm	per person
Offices	15	"	"
Offices (smoking)	30	"	"
Restaurants	15	"	"
Theaters	7½	"	"
Broker's Room	30	"	"
Cocktail Lounge	25	"	"

The quantity of outside air finally established is deducted from the total air quantity supplied (which has previously been determined from the sensible heat load); the balance is recirculated air. Thus, if there are 200 persons occupying a restaurant, 200 × 15 = 3000 cu ft of outside air would be supplied. Since the total cubic feet of air supplied per minute was established as 9333 cfm in our previous example, the amount recirculated would be 9333 cfm — 3000 cfm = 6333 cfm recirculated.

This air must be cooled to the dewpoint temperature, which was previously established at 56 F. The outside air condition is taken as 95 F dry bulb and 75 F wet bulb (the total heat is 38.46 Btu per lb) and the total heat is 23.77 Btu per lb at the 56 F dewpoint. The volume of air at the 56 F dewpoint is 13.19 cu ft per lb. The heat to be extracted from the outside air is:

$$\frac{3000 \times (38.46\text{-}23.77) \times 60}{13.19}$$

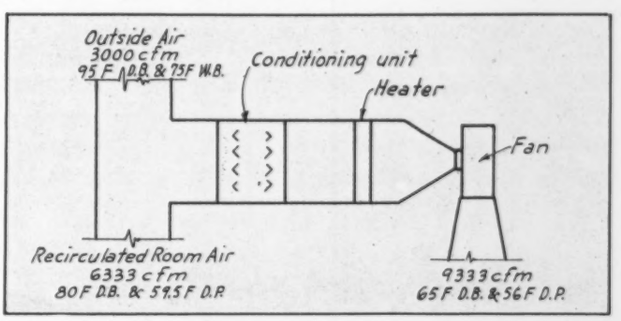
$$= \frac{3000 \times 14.69 \times 60}{13.19}$$

$$= 200{,}488 \text{ Btu per hr}$$

Refrigeration Load

The refrigeration load consists of the total of the sensible heat, latent heat, and outside air heat, and can be easily calculated from the quantity of air supplied.

The requirements of our problem are to cool the air to a leaving condition of 56 F saturated and to reheat to a dry bulb of 65 F, the total quantity of air supplied being 9333 cfm.



Cooling the 6333 cfm of recirculated air from 80 F D. B. and 59.5 F dewpoint to 56 F dewpoint requires the extraction of 3622 Btu per min.:

$$\frac{6333 \text{ cfm}}{13.19} = 480 \text{ lb per min}$$

At room wet bulb (66.7 F) total heat of air = 31.276 Btu per lb
At apparatus dewpoint (56 F) total heat of air = 23.770 Btu per lb

Difference = 7.506 Btu per lb
480 lb × 7.506 = 3622 Btu per min

Cooling the 3000 cfm of outside air from 95 F dry bulb and 75 F wet bulb to 56 F dewpoint requires the extraction of 3335 Btu per min.:

$$\frac{3000 \text{ cfm}}{13.19} = 227 \text{ lb per min}$$

At outside wet bulb (75 F) total heat of air = 38.46 Btu per lb

*Engineer, Carrier Corp.

Part 5 of a series. Part 1 was published in May, pp. 281-284; Part 2 in June, pp. 367-368; Part 3 in July, pp. 415-417; and Part 4 in August, pp. 488-491.

At apparatus dewpoint (56 F) total heat of air = 23.77 Btu per lb
Difference = 14.69 Btu per lb
227 lb × 14.69 = 3335 Btu per min

Total Refrigeration Load

Recirculated Air ..... 3622 Btu per min
Outside Air ..... 3335 Btu per min

Total ..... 6957 Btu per min

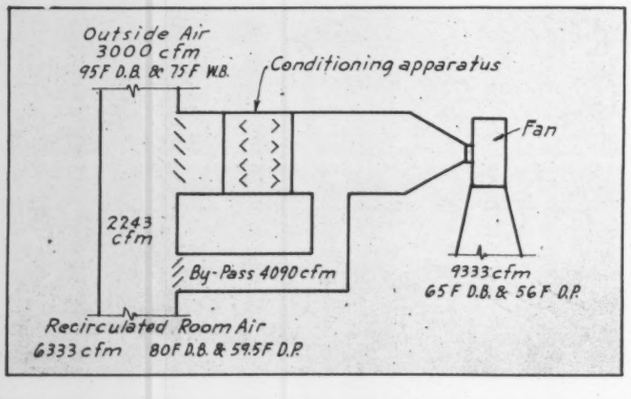
6957 × 60 = 417,420 Btu per hr

$$\frac{417{,}420 \text{ Btu per hr}}{12{,}000} = 35 \text{ tons of refrigeration required.}$$

From the nature of the air conditioning apparatus it can be seen that the air leaving is at a saturated condition of 56 F, which is the necessary dewpoint to which the air must be reduced if proper dehumidification is to be accomplished. It is necessary to reheat the air with steam or some other medium to the correct entering dry bulb temperature of 65 F to avoid overcooling. When the air conditioning requirements are lessened due to lower outdoor temperature, fewer people, less lighting, etc. the air must be further heated from the 56 F saturated condition even to as high as 70 or 75 F to avoid overcooling under reduced load conditions.

The Bypass System

In the bypass† system a portion of the recirculated air is mixed with the outside air which has been passed through the conditioner together with the remainder of the recirculated air; the quantity of recirculated air being automatically controlled warms the 56 F saturated air leaving the dehumidifier to the correct dry bulb temperature. Since only the portion of recirculated air that passes through cooling apparatus is refrigerated (the balance being bypassed around the cooling apparatus), the refrigeration load is reduced proportionally. In the problem under consideration the quantity of air bypassed is 4090 cfm.



To determine the approximate quantity of air bypassed under maximum conditions the following method can be used:

$$AX + BY = C$$
$$DX + BY = E$$

where A = Room dry bulb temperature.
B = Corrected dewpoint temperature.
C = Entering dry bulb temperature.
D = Room dewpoint temperature.
E = Calculated entering air dewpoint temperature.
X = Per cent of air bypassed air conditioner.
Y = Per cent of air passed through air conditioner.

In the problem under consideration, $A = 80$ F, B is unknown, $C = 65$ F, $D = 59.5$ F, and $E = 56$ F.

$$80X + BY = 65$$
$$59.5X + BY = 56$$

Subtracting, $20.5X = 9$
$X = 9 \div 20.5 = 0.438 = 43.8$ per cent
$Y = 100 - 43.8 = 56.2$ per cent

To determine the new dewpoint temperature B, substitution can be made in one of the formulas.

$$80X + BY = 65$$
$$(80)\ (0.438) + (B)\ (0.562) = 65$$
$$35.1 + 0.562B = 65$$
$$0.562B = 29.9$$

$$B = \frac{29.9}{0.562}$$

$$B = 53.2 \text{ F}$$

Thus the dewpoint must be slightly reduced from 56 to 53.2 F when using the bypass system. The quantity X of recirculated air bypassed is:

0.438 × 9333 cfm = 4090 cfm
6333 − 4090 = 2243 cfm passes through the conditioner.

The refrigeration load is now corrected to allow 3000 cfm of outside air to pass through the conditioner and 2243 cfm of recirculated air to pass through the conditioner, all of which is cooled to 53.2 F saturated.

At room wet bulb (66.7 F) total heat = 31.276 Btu per lb
At apparatus dewpoint (53.2 F) total heat = 22.680 Btu per lb

8.596 Btu per lb

$$\frac{2243}{13.09} = 171.6 \text{ lb per min}$$

171.6 × 8.596 = 1472 Btu per min

At outside wet bulb (75 F) total heat = 38.46 Btu per lb
At apparatus dewpoint (53.2 F) total heat = 22.68 Btu per lb

15.78 Btu per lb

$$\frac{3000}{13.09} = 229 \text{ lb per min}$$

229 × 15.78 = 3620 Btu per min

Total Refrigeration Load

Recirculated air = 1472 Btu per min
Outside air = 3620 Btu per min

5092 Btu per min

5092 × 60 = 305,000 Btu per hr
305,000 ÷ 12,000 = 25.5 tons

Thus by the use of the bypass the required refrigeration capacity is reduced from 35 to 25.5 tons and the need for reheating with steam is eliminated.

[To be concluded]

†Information on patent license for use of this system may be had from Auditorium Conditioning Corp., 17 E. 42nd St., New York, N. Y.

The Owner's View of Air Conditioning

Walgreen Drug Co.—Drug Stores

By **Arthur Behrstock***

Some 125 of approximately 525 Walgreen drug stores have cooling equipment, and the company's total air conditioning investment represents between one-half to three-quarters of a million dollars. . . . . What Walgreen's have learned about air conditioning, what they think about it, is briefly reviewed

IN the days when bicycles were built for two, Charles R. Walgreen was a young pharmacist who set out to blaze a new trail, embarking, in his little drug store on the south side of Chicago, on a series of improvements that were revolutionary in their day. He was one of those who started the tradition of the clean drug store; he let in more light; he put more space between the aisles.

The Walgreen company artificially cooled its first store seven or eight years ago, experimenting with two stores almost simultaneously—one in the loop in Chicago, the other in Louisville, Ky. During the ensuing years, it experimented with new types of equipment as they made their appearance; the process of experimentation still continues.

Today, about 125 of approximately 525 stores are air conditioned, and the company's total air conditioning investment, distributed in almost every section of the country, represents between one-half and three-quarters of a million dollars. The heaviest concentration of conditioned stores is in the south and southwest, where, in some parts, the equipment runs almost the entire year around. Cooling equipment is installed in almost every new store where the climate demands it, while in the south all new stores are being air conditioned.

Many of the reasons for the company's extensive conditioning of the stores are obvious; others are not so obvious. In a hot climate, the patron obviously enjoys being cool. But this is not the only advantage. It has been found that the customer receives better service from a waitress or soda man who is cool also; a perspiring waitress is inclined to be irascible and sharp. It has also been found that candies are preserved better and that, because of the filtered air, goods are cleaner in a conditioned store.

Problems in Cooling the Drug Store

There are a number of air conditioning problems peculiar to the drug store. The biggest is to crowd the equipment into the small space available. The heat load in drug stores, due to the moisture, the fountain and the cooking equipment, is high. Large numbers of people in a small space, the intensity of the lighting and numerous heating and power appliances do not make the problem any easier. There are the additional problems of roofs directly above the store and of front entrance doors that must be kept open in order to maintain an inviting appearance.

To cope with these problems, Walgreen's has its own architectural and engineering departments. Roofs are insulated when they are directly above stores. To meet the door problem, a special lighter door is installed which is not as formidable and uninviting as the heavier door. To obtain natural sunlight, while keeping the hottest rays of the sun out, venetian blinds are often used. It has been found necessary to segregate cooking and fountain equipment under ventilator hoods to keep heat and odors from diffusing into the cool air.

Much experimenting lies behind the system of air conditioning now prevalent in Walgreen stores. Different kinds of equipment have been used—coolers hung in the store were tried; then units with compressors in them.

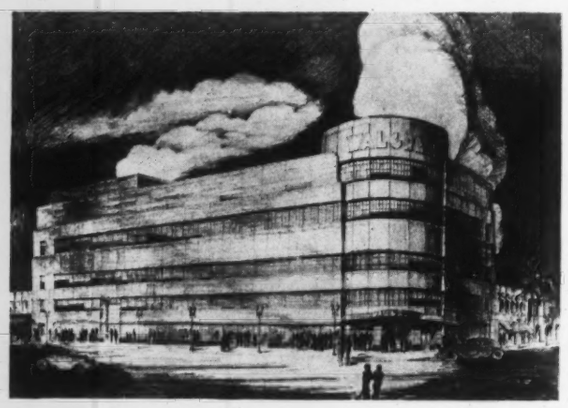
Unitary equipment with "Freon" compressors and direct expansion coils is now used almost exclusively, with the air blown through duct work. Equipment is installed outside of the sales premises. In some of the smaller stores, small direct cooling units are suspended directly in the store or placed on the floor. Installation of the duct work presents an engineering problem in the older and smaller stores. In places where wells or other sources supply extremely cold water, this water is used for complete or partial cooling. Where water is extremely expensive, evaporative condensers are used, which cut the water bill to about 10 per cent of what it might be. This equipment is used in about 35 per cent of the stores.

The now antedated ideal of summer air conditioning was to keep the temperature at 70 degrees, to keep it, that is, at inside winter heat. This "ideal" Walgreen's has given up, because it is now generally recognized that sudden differences in temperature are dangerous and unhealthy. Thus, a customer stepping into a 70 degree store, when the temperature outside is 100, would step into a 30 degree difference in climate; the same process would be repeated on leaving the store. The company now tries to keep a 12 degree difference between outside and inside, when outside temperatures are around 95; when the outside temperature is less, the range between the two is diminished, and when outside temperatures go above 95 degrees, the range is increased. At the same time, the relative humidity in the store is kept between 50 and 55 per cent. Extreme care is taken to obtain even distribution of air throughout the store, without excessive drafts.

*Walgreen Drug Co.

New Miami Store Has Many Features

Although the average cost of operation of air conditioning equipment in one store for a season runs between $500 and $1000, Walgreen's new Miami store, which will be one of the largest and most complete drug stores in the world, will hardly be able to come within these figures. The store—the equipment and building for which cost nearly $1,000,000—will be a modernistic structure of five floors. It represents in many ways the



Walgreen's new Miami store

culmination of the company's experimenting in the shape of things to come. It is also a continuation of the adventuring spirit that guided Charles R. Walgreen during his early corner drug days, for the new store will be an adventure in many ways. Possessing an 80 ft streamlined fountain and capable of serving 700 persons at one time from the fountain and its many kitchens, the store will be the last word in technical design. A network of interior telephones in the dining rooms will be connected with the kitchens, so that waitresses may telephone their orders. A public address system, with 30 amplifiers distributed throughout the building, will be operated from a glass enclosed broadcasting studio, so that customers may watch the proceedings at the microphone.

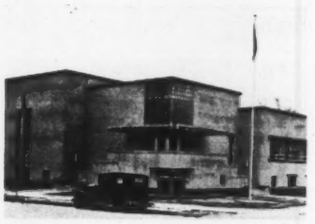
The air conditioning equipment will be of similar proportions. Costing approximately $50,000, it will have a capacity of about 100 tons of refrigeration. To avoid cold drafts, the cooled air will be blown up from the sides of the store, the second and third stories of which will be a balcony and mezzanine. The basement—the first merchandising basement in Miami—will be cooled by a separate air conditioning unit. Because there is a good deal of salt in the well water, admiralty metal will be used for the shell and tube condensers in order to withstand the action of the salt.

Advantages of Drug Store Cooling

The company has found it difficult to measure, with scientific precision, the exact effect of air conditioning on sales volume and profits. There are too many variable factors involved for such precision: The business cycle may go up or down; a new theater may open in the region of a store; fresh competition may arise. However, certain results can be noted. In the first survey taken in a Chicago store in July, 1934, a total increase in volume of 3.5 per cent was noted (8 per cent in fountain business, and 2 per cent in all other sales). Subsequent installations in stores in various cities show an increase in volume of from 2 to 7 per cent for the year. But, again, not all stores come out even approximately alike. In a large midwestern city, two stores that were air conditioned did not show as good results as four that were not, but outside factors may account for these figures.

There are many reasons given by the company for air conditioning its stores—some of these reasons varying from locality to locality and from store to store—but three take precedence, perhaps, over all others—the comfort of the customers; the health and, consequently, the "temperament" of employees; and the greater cleanliness of merchandise.

School Administration Building Is Air Conditioned



THE school administration building at Fresno, Calif., has recently been equipped for year 'round air conditioning; the building comprises offices for school officials, a library, an auditorium, and a warehouse for school supplies.

The air conditioning equipment consists of individual units placed in various parts of the building under thermostatic control to control just certain parts of the building. For example the auditorium is equipped with an individual unit under thermostatic control, as is the visual education room. The units were made up on the job, using standard evaporators, and steam coils.

The automatic control consists of the necessary thermostats for controlling the motorized steam valves, and electric operated solenoid valves controlling the liquid refrigerant to the evaporators for the various units. The condensing unit is controlled automatically by pressure controllers, so that the compressor starts and stops depending upon the back pressure of the system.

Edward B. Ward Co. made the installation.

A Simplified Method for Solving

Air Washer Problems

By Dr. J. R. Zwickl*

A simple and easily understandable graphical method which lends itself to solving problems that come up in connection with spray water air coolers and air washers is discussed. The method is based on the heat balance between air and water. It involves the use of only one curve, which can be laid out by anyone from data in the *A. S. H. V. E. Guide*. . . . . All other lines are straight ones

WHEN cooling air by spray type coolers or air washers problems arise, some of which can be solved directly and others only by trial and error methods. For solving these latter problems, William Goodman devised his "spray water-air chart" and published an article about its use in HEATING, PIPING AND AIR CONDITIONING.[1]

Some time ago the writer had to solve similar problems, and worked out a simple and direct graphical method which lends itself so well to the purpose that it seems worthwhile to discuss this method and its use. The problems to be solved being about the same, it is quite natural that the present article runs parallel to Mr. Goodman's, the difference being mainly in the method used.

For this discussion, an air washer is considered merely as a heat exchange apparatus; the air to be cooled is passed through an insulated chamber, where it comes in intimate contact with a finely divided cold water spray. During this process the air cools and the water is heated. As no other heat is introduced, it is evident that all the heat surrendered by the air is picked up by the water spray. This heat balance is expressed in the equation:

$$W\ (t_2 - t_1) = A\ (h_1 - h_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where

W = weight of water sprayed into the washer, lb per minute,
t_1 = initial water temperature, F,
t_2 = final water temperature, F,
A = weight of air passing through washer, lb per minute,
h_1 = total heat of air at the initial wet bulb temperature T_1, Btu per lb,
h_2 = total heat of air at the final wet bulb temperature T_2, Btu per lb.

If we introduce $W_1 = W/A$ as the amount of water in pounds that is sprayed into the washer for every pound of air, we can write Equation [1] as:

$$W_1\ (t_2 - t_1) = (h_1 - h_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [2]$$

In the following, W_1 will be called the "water rate." Equations [1] and [2] hold true, whether the air washer is 100 per cent efficient or less; the difference is only that in the 100 per cent efficient (or "perfect") air washer, the final wet bulb temperature of the air is the same as the final temperature of the water spray ($t_2 = T_2$), whereas in the less efficient washer $t'_2 < T'_2$. The temperature difference $\Delta = (T'_2 - t'_2)$ depends on the construction of the air washer and also on the water rate W_1, which is evident, since the more water is sprayed for every pound of air, the larger is the surface of the water spray in suspension and therefore the final air temperature will approach closer to the final water temperature. In discussing the different problems, a perfect air washer is assumed, *i. e.* air and water in contact with each other reach one and the same final temperature in each spray zone.

Air washers are usually made in two types, single stage and double stage. Single stage washers have only one bank of nozzles, and all nozzles are supplied equally with water of the same initial temperature. Thus this type of washer has only one spray zone. Two stage washers have two banks of nozzles; *i. e.* two spray zones. If the water is first supplied to the second bank of nozzles (in the direction of the air flow) and then to the first bank of nozzles, we have a counterflow air washer. If both banks are supplied with water of the same initial temperature, we have a two stage split cooler. Each of these types requires a somewhat different method for solving the problems.

Solving Single Stage Air Washer Problems

The total heat h of an air-water vapor mixture is definitely determined by its wet bulb temperature T. If we know T, we know also the corresponding value of h. Thus in Equation [2] the value of $(h_1 - h_2)$ can also be expressed in relation to T the wet bulb temperature. The easiest way to do this is to plot the curve $h = f(T)$, as is done in Fig. 1. The values of h given on this curve were taken from the 1936 edition of the *A. S. H. V. E. Guide*. The curve is plotted with the temperature as ordinate and h as abscissa, measured to the left from its zero point, which is not shown. If we locate on this curve (to be called the "air curve") two wet bulb temperatures T_1 and T_2, then the horizontal distance between these two points is $(h_1 - h_2)$, which is the heat given off by each pound of air while its wet bulb temperature decreases from T_1 to T_2. In other words, when the air cools, its condition changes along the air curve, or the $h = f(T)$ line.

The heat picked up by the water causes a rise in its temperature. If Q designates the heat (in Btu) which is lost by each pound of air during its passage through the spray zone and which is taken up by the corresponding W_1 lb of water sprayed into the washer for each pound of air, then we can write: $Q = W_1\ (t_2 - t_1)$. In

*Consulting Engineer, Baldwin-Southwark Corp.

[1] HEATING, PIPING AND AIR CONDITIONING, May, 1935, pp. 232-235, and August, 1935, pp. 382-385.

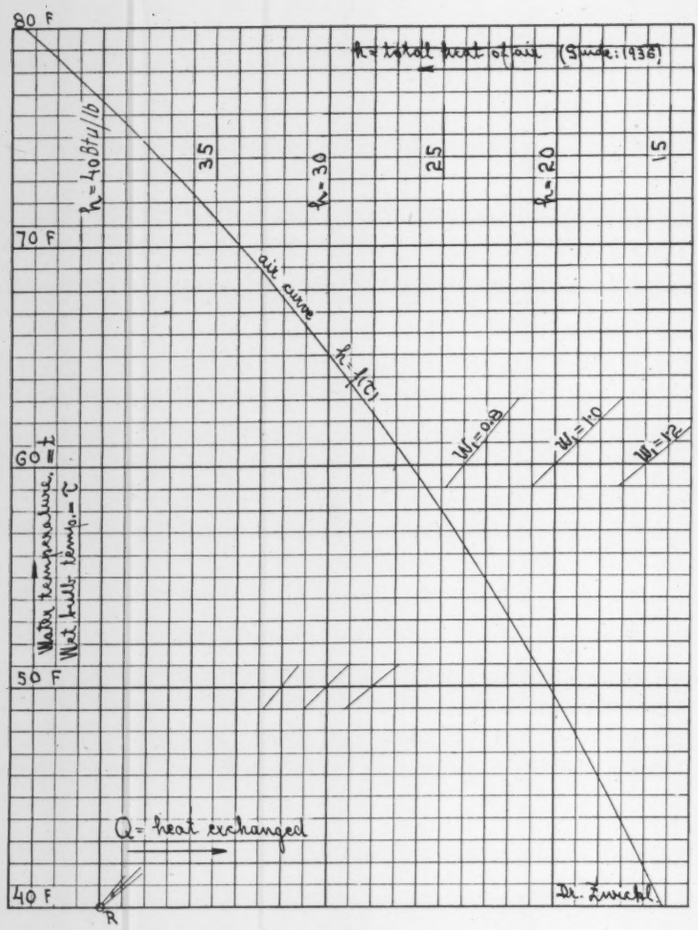


Fig. 1—As the total heat h of an air-water vapor mixture is a function of the wet bulb temperature T, the curve $h = f(T)$ is plotted. When the air cools, its condition changes along this line, called the "air curve"

a "Q—t" diagram this equation is represented by a straight line, which has the slope $tg\alpha = W_1$ toward the t axis of the diagram.[2] Thus we can write $Q = W_1 (t_2 - t_1) = tg\alpha\ (t_2 - t_1)$ and this line can easily be drawn into Fig. 1. The scale for the two axes of the diagram being the same, *i. e.* the scale length for 1 Btu and the scale length for 1 F being the same, the water line for $W_1 = 1$ has a slope of 45 degrees. For any other value of W_1 the corresponding "water line" can be laid down by figuring out Q_x for a 10 or 20 F rise in temperature and locating this point on the corresponding temperature line at the distance of Q_x units to the right from the corresponding $Q = 0$ line. This latter one is a vertical line and its location on the diagram is set by T_1, the initial wet bulb temperature of the air: the $Q = 0$ line crosses the air curve at the initial wet bulb temperature T_1. In Fig. 2 the relation given by Equation [2] is laid down graphically. Point A is located where T_1 crosses the air curve. Through A we draw a vertical line (the $Q = 0$ line) and where this line cuts the t_1 line (initial water temperature) there is located point B. From B we draw the water line under the slope $tg\alpha = W_1$ as explained above. A is the starting point for the cooling of the air, whereas B is the starting point for the heating of the water and when they reach the common point C, the air has given off just as much heat as the water has picked up: Thus the point C represents the final condition, for which $T_2 = t_2$ as supposed.

[2] "$tg\alpha$" means the tangent of the angle alpha and—see Fig. 2—means the ratio $Q/(t_2 - t_1) = \dfrac{h_1 - h_2}{t_2 - t_1} = W_1$.

Fig. 2 enables us to solve all the problems that might come up in connection with a single stage air washer. In Equation [2] we have five variables; any three of them may be given, the other two are to be found.

Problem 1. Given T_1 and T_2 for the initial and final wet bulb temperature of the air, and t_1, the temperature of the available spray water. What is the water rate W_1?

This problem can be solved directly and also graphically. Read from the air curve, Fig. 1, the heat contents h_1 and h_2 corresponding to T_1 and T_2. Since it is assumed that $t_2 = T_2$, we know also t_2, so that from Equation [2] we find $W_1 = (h_1 - h_2) \div (t_2 - t_1)$.

Graphically we can do it as follows: Having plotted the "air curve," *i.e.*, $h = f(T)$ on a large enough cross-section paper in the way as shown in Fig. 1 (with equal scale length for ordinate and abscissa), we then locate on this curve point A at the crossing with the T_1 wet bulb temperature and point C at the crossing with the T_2 wet bulb temperature, as shown in Fig. 2. From point A we draw a vertical line (the $Q = 0$ line) and where this crosses the t_1 temperature line (initial water temperature) there is located point B, where the "water line" BC starts. From the angle α of this line, or rather from its slope we can find $W_1 = tg\alpha$ either by actually calculating it as $tg\alpha = (h_1 - h_2)/(t_2 - t_1) = W_1$, which data can be read from the diagram; or graphically by drawing a line parallel to the water line through point R, which—as shown in Fig. 1—is located at the crossing of the $t = 40$ F line with the $h = 40$ Btu per lb line. This parallel line cuts the $t = 50$ F line at a

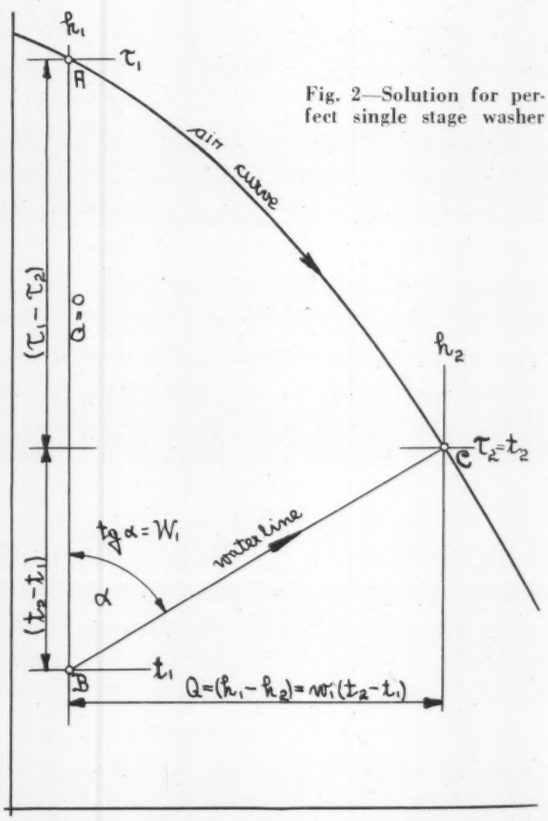


Fig. 2—Solution for perfect single stage washer

horizontal distance of 10 W_1 Btu to the right of the $h = 40$ Btu reference line, so that we can read the value of 10 W_1 (or W_1) directly from the diagram without any figuring. (In a reversed way we can use this latter procedure also for setting the slope $tg\alpha = W_1$ of the water line).

Problem 2. Given T_1, t_1 and W_1. Find the final air and water temperature $T_2 = t_2$.

This problem can be solved only graphically as shown in Fig. 2. Locate points A and B corresponding to the given temperature T_1 and t_1. Draw from B the water line with the slope $tg\alpha = W_1$. This cuts the air curve in point C, which fixes the temperature $T_2 = t_2$. Point C being fixed, we can read from the air curve $(h_1 - h_2)$, the heat surrendered by each pound of the air to be cooled.

Problem 3. Given T_1 and T_2, also the water rate W_1. What must be the initial water temperature t_1?

This problem can be solved directly and graphically. Read from the air curve h_1 and h_2. Find from Equation [2]:

$$(t_2 - t_1) = (h_1 - h_2) \div W_1$$
$$\text{and } t_1 = T_2 - (t_2 - t_1)$$

Graphically: Locate points A and C on the air curve. Draw through A the vertical line $Q = 0$. Draw through C the water line with the slope $tg\alpha = W_1$. This water line will cut the $Q = 0$ line in point B, which fixes the initial water temperature t_1.

Problem 4. Given t_1, W_1 and $T_2 = t_2$. What can be the highest T_1 (initial wet bulb temperature)?

This problem can be solved directly and graphically. From Equation [2] figure out:

$$(h_1 - h_2) = W_1 (t_2 - t_1)$$
$$\text{and } h_1 = h_2 + (h_1 - h_2)$$

Finally find T_1 to h_1 from air curve.

Graphically: Locate point C on air curve to T_2. Draw through C the water line with the slope $tg\alpha = W_1$. This line cuts the t_1 temperature line in point B. Erect in B the vertical line $Q = 0$, which finally cuts the air curve in point A, which fixes the initial wet bulb temperature T_1.

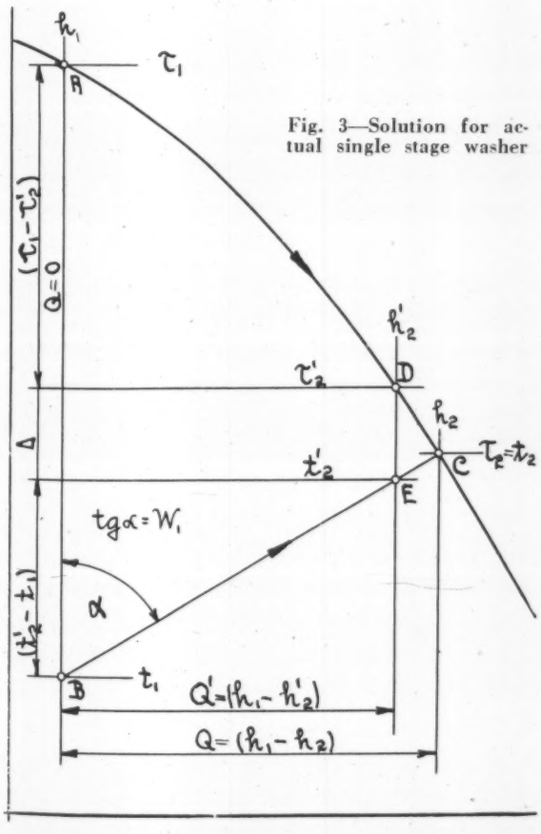


Fig. 3—Solution for actual single stage washer

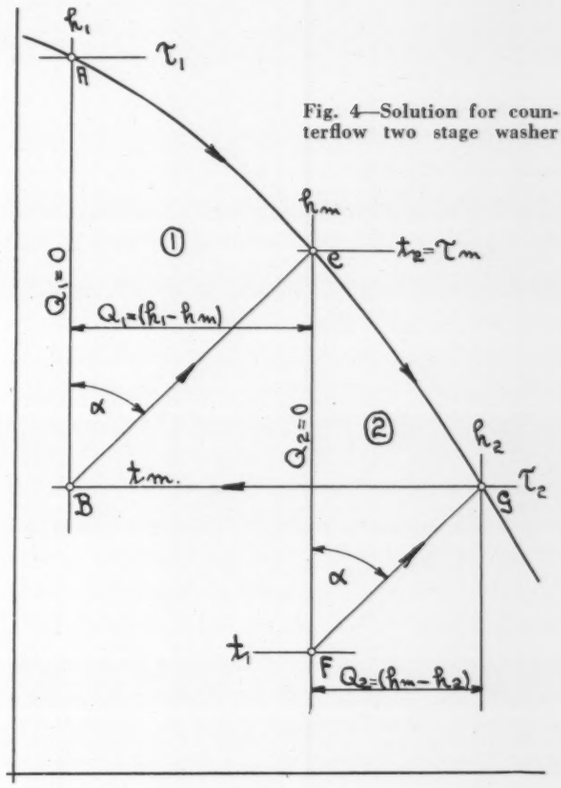


Fig. 4—Solution for counterflow two stage washer

Up till now we have considered a perfect air washer in which $T_2 = t_2$, *i. e.* the final wet bulb temperature of the air and the final temperature of the spray water are the same. Such an air washer does not exist and in reality these two temperatures only approach each other without ever being equal. Referring to Fig. 3, we see that the final wet bulb temperature $T'_2 > T_2$, also $t'_2 < t_2$. The air cools only to point D (on the air curve) and the water heats only to point E (on the water line). The difference $\Delta = (T'_2 - t'_2)$ is characteristic for the washer and depends on the construction and the water rate W_1. The heat exchanged in the process is only $Q' = (h_1 - h'_2)$ instead of $Q = (h_1 - h_2)$ as for the perfect washer. If we call the ratio $Q'/Q = e$, the efficiency of the washer, we can find $Q = Q'/e$ and knowing Q we can locate the point C on the air curve and handle the problem as for a perfect washer. The numerical value of Δ can be expressed even by an equation containing e, the efficiency of the washer, and W_1, the water rate, but this would complicate the solution unduly.

Solving Two Stage Air Washer Problems

Counterflow two stage washer

This type of washer is chacterized by having the same water first passed through the second bank of nozzles (colder spray zone) and after that, without any intermediate cooling, through the first bank of nozzles (warm spray zone). Note: "First" and "second" is to be taken with reference to the direction of the air flow through the washer.

From the above it follows that the water rate W_1 is the same for each spray zone. Assuming again a perfect washer, we take the final air and water temperatures as equal after each spray zone. Such a counterflow air

washer can be regarded as two single stage washers in series so that the first one receives its spray water after it has passed through the second washer; this again receives the air after it passed through the first washer. With reference to Fig. 4 we can see that the intermediate water temperature t_m (after the second spray zone) is the same as the final air temperature T_2.

First stage: The air enters with T_1 wet bulb temperature (point A, on the air curve) and cools to $T_m = t_2$ temperature (point C). The water enters with t_m temperature and leaves with t_2 temperature. The heat balance for the first stage will be:

$$Q_1 = (h_1 - h_m) = W_1 (t_2 - t_m) \quad \text{[2a]}$$

where $W_1 = tg\alpha$ as before.

Second stage: The air enters with T_m wet bulb temperature (point C on air curve) and cools to $T_2 = t_m$ final wet bulb temperature (point G). The water enters with the initial temperature t_1 and leaves with $t_m = T_2$ temperature. Thus the heat balance will be:

$$Q_2 = (h_m - h_2) = W_1 (t_m - t_1) \quad \text{[2b]}$$

Adding Equation [2a] to [2b] results in Equation [2], when it is applied to the whole washer as one unit.

When scrutinizing Fig. 4, we see that point A and point B are again, as before, on the same vertical line ($Q_1 = 0$), which is fixed by the location of A on the air curve at the T_1 temperature. Analogously, point C and point F too are on one vertical line ($Q_2 = 0$), which is fixed by the location of C on the air curve at the temperature $T_m = t_2$. Furthermore point B and point G are on the same horizontal line fixed by the temperature $t_m = T_2$. We see also that point C on the air curve represents the end condition of the air for the first stage and the entrance condition for the second stage.

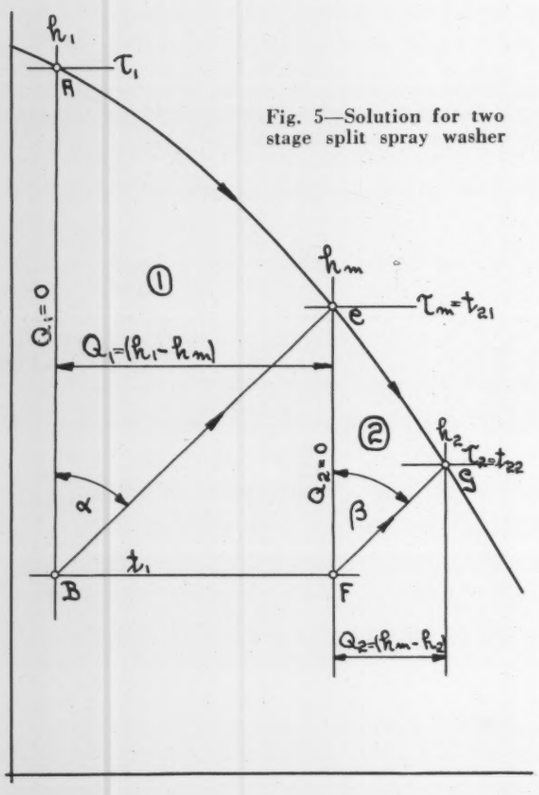


Fig. 5—Solution for two stage split spray washer

By means of the above relations illustrated graphically in Fig. 4, we are able to solve problems pertaining to the counterflow two stage air washer.

Problem 5. Given T_1 and T_2, and the water rate W_1. Find the initial water temperature t_1 required, also the final water temperature t_2.

The wet bulb T_1 fixes A on the air curve, whereas T_2 fixes point G. Since $t_m = T_2$, the horizontal line for t_m is hereby fixed. Draw through A the vertical line $Q_1 = 0$; this line meets the t_m horizontal line in point B, which is the starting point of the water line for the first stage. This water line can be laid in, since we know W_1, the water rate, and $tg\alpha = W_1$. In this way we find point C on the air curve, which fixes T_m, the intermediate air temperature, and also $t_2 = T_m$, the final water temperature. Point C represents the initial condition of the air for the second stage, therefore it fixes the location of the $Q_2 = 0$ line; this vertical line can be drawn in. If we lay through point G the water line for the second stage (it runs parallel with the first one, since W_1 is the same for both stages), this water line meets the $Q_2 = 0$ line in point F and so fixes the required initial water temperature t_1, which was the problem.

Problem 6. Given t_1 and W_1, also T_2. What is the highest permissible initial wet bulb temperature T_1 for the entering air?

The final wet bulb $T_2 = t_m$ fixes point G on the air curve, also the t_m horizontal temperature line. Lay through point G the water line for the second stage with the slope $tg\alpha = W_1$ which is given. This water line cuts the t_1 temperature line in point F, which is the lower end of the $Q_2 = 0$ line. Erect this line and so find point C on the air curve. Draw through C the water line for the first stage (parallel to the other water line) and find B on the t_m horizontal temperature line. Erect in B the vertical line $Q_1 = 0$, which cuts the air curve in point A at the wet bulb T_1, which was to be found. The final water temperature t_2 is fixed by the location of point C on the air curve.

Problem 7. Given T_1 and T_2 for the air, and the initial water temperature t_1. Find the water rate W_1 required, and also the final water temperature t_2.

Solving this problem involves some trial and error, but it is done easily. The wet bulbs T_1 and T_2 locate points A and G on the air curve. Drop vertical line $Q_1 = 0$ from point A and draw through G the horizontal line $t_m = T_2$. This line locates B on the $Q_1 = 0$ line. Draw in also the line t_1. Points B and G each are located on a water line. Draw in these lines parallel to each other with an assumed slope $tg\alpha$, and find a point C_x on the air curve and a point F_x on the t_1 line. Modify the slope ($tg\alpha$) until F_x and C_x fall on the same vertical line (the $Q_2 = 0$ line). If this requirement is fulfilled, the slope of the water lines is correct and from it we can find $W_1 = tg\alpha$ as explained earlier in this discussion.

Having found the correct W_1, we find the final water temperature t_2 as fixed by the correct point C on the air curve. By using two triangles the correct slope for the water lines is found very easily; three or four trials suffice.

Problem 8. Given t_1 and W_1 for the water and T_1 for the air. To what final T_2 wet bulb temperature can the air be cooled?

The solution of this problem is very similar to that of the previous one. The wet bulb T_1 locates point A on the air curve. From A we drop the vertical line $Q_1 = 0$. We draw in also the t_1 temperature line. Since W_1 is given, we can lay out the slope of the water lines as $tg\alpha = W_1$. By assuming any intermediate water temperature t_m, we fix the corresponding point B_x on the $Q_1 = 0$ line and G_x on the air curve. Draw from B_x and G_x the water lines with the given slope; they cut out a corresponding C_x point on the air curve and an F_x point on the t_1 temperature line. Since the correct C_x and F_x points must be located on the same $Q_2 = 0$ vertical line, modify t_m, *i. e.*, B_x and G_x, until this requirement is fulfilled. The correctly located G_x point fixes T_2, the required final wet bulb of the air.

We can also assume the location of the $Q_2 = 0$ line with the

[*Concluded on page* 563]

The Romans Had a Technique for It

Early Manufacture of Lead Pipe

By F. E. Wertheim

ENGINEERING history, that most fascinating study, reveals over and over how extensively "the ancients stole *our* ideas."

In the investigation of water supply, we find that Rome, in the first century, not only had a remarkable system of aqueducts, mains and storage reservoirs, but also had a separate group of officials to administer this service. Water from springs and lakes, located at distances as great as 60 miles, was conveyed by gravity flow to storage tanks (called *castellae*) built at convenient points within the city; from which, again by gravity flow, it was distributed by lead mains and piping.

The purpose of this article is to outline the methods employed to make lead pipe, dating from about A. D. 97 to the time of Robert Brocke, who invented moulded, cast-round, lead pipe in 1539.[1]

Much lead pipe of ancient origin has been found in Rome, Pompeii, and Bath, England. For the latter pipes, the lead was obtained from the Roman mines in the Mendip Hills, and contained a considerable proportion of silver, although this fact is not advanced as a reason for the more economical use of material at Bath, where

Fig. 1—Triangular and rectangular pipe sections

the pipe wall thickness was only 0.25 in., while that at Pompeii measured 0.36 in. The Romans designated pipe sizes with relation to girth; the diameters ranged from 0.75 in. to about 24 in., and occasionally reached 27 in.

Roman pipe was made by first casting flat sheets of lead in a mould with a smooth sand floor. These sheets were cut into strips about 10 ft long, with width corresponding to the desired circumference, and were then bent and hammered into pipe form. Opinions differ concerning the shape of the pipe. Fig. 1, taken at Bath, shows triangular and rectangular sections, which, some maintain, is the way the piping was made; while others assert that the pipes were crushed to these shapes by the weight of debris. Another theory is that the strips were formed into a pear shape by beating them around a wooden core until the longitudinal edges met. Fig. 2 shows a number of sections cut from such pipes.

The raised seams were fused as follows: The tubes were filled with sand and then buried in sand in a manner to form a shallow trough at the seam. It is probable that the edges were preheated by drawing a red-hot iron over them, after which molten lead, at a temperature at least 50 degrees above the melting point, was poured along the trough, forming the raised longitudinal seam. Some of this pipe was very neatly made, with the excess

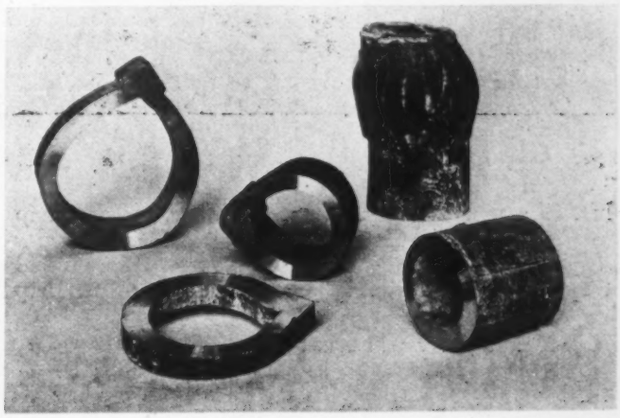


Fig. 2—Sections cut from early lead pipe

lead wiped or scraped off before solidification. Fig. 3 is a full-size photo of a macro-etched section of the pipe which is seen at the extreme left of Fig. 2. The long crystals at the joint indicate that cooling was very slow due to the presence of a relatively large mass of poured metal.

Joints between lengths were made *in situ* by filling the pipe ends with sand, butting them together and building a sand mould around them to cast a collar which would join the ends. Molten lead was poured into the collar cavity and overflowed a small weir until the ends fused together; a fact which was proved by probing with a wooden splinter until penetration to the sand core was possible. This method of joining lengths of pipe persisted until the middle of the 16th century—many years after the introduction of the wiped joint. In Fig. 2 the upper righthand picture shows an early wiped joint, which has been "overcast" by rubbing a hot iron longitudinally over the joint. Although this resulted in a gashed surface, pores in the metal were effectively closed.

Crude as these methods may now seem, recent hydrostatic tests of some of this old piping have shown that at 112 lb per sq in. the pear shape changed to a circular section and that failure did not occur until the pressure

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[1] Any reader interested in more extensive or detailed information is referred to "Frontinus and the Water Supply of the City of Rome," by Clemens Herschel, 1913, and Smith's Dictionary of Greek and Roman Antiquities, revised by Wayte and Marindin, 1890-91.

Air Conditioning in the Tropics Presents New Engineering Problems

By J. Neal Clemmer*

Mr. Clemmer recently returned from a business trip to the Far East. He flew by Pan-American Clipper ships to Manila, Philippine Islands, where he studied tropical air conditioning problems. His return trip included visits to China, Japan, and Honolulu . . . . He points out here some of the problems of air conditioning in the tropics, and how air conditioning practice there varies from that in the United States

THE engineer who has been accustomed to designing air conditioning installations in the U.S.A. may well expect some surprises when he packs up and goes globe-trotting. He will discover any number of factors that enter into every tropical air conditioning installation that would never in all probability have reached his attention if he had not gone out to the scene of action.

Illustrating my point further, I found out shortly after arriving in Manila that we had to consider the human equation from a new light in designing air conditioning installations there. Inhabitants of tropical countries are accustomed to warm weather the year around, year in and year out. Consequently, the dry bulb temperatures which are demanded by people in this country as being standards of "comfort cooling" are far too cold for Philippine residents. Numerous complaints were voiced to me about the drastic temperature differences maintained by most air conditioned establishments (theaters particularly). It was too chilly, they thought, although most American visitors there would feel that the rooms were just comfortable. A temperature of 80 F dry bulb and 55 per cent relative humidity seems to be a minimum design condition in the Philippines with an outside dry bulb of 92 F. An even closer temperature difference, no doubt, would be pronounced comfortable by the majority of people.

The Philippines, consisting of 7083 islands, stretch 1152 statute miles from north to south and as a result, possess wide variations of tropical and semi-tropical climates. Manila, however, located on the largest island (Luzon) is the most important city and serves as an excellent laboratory for the study of tropical air conditioning problems.

April, May and June are the hot months there, with temperatures usually at a mean average of about 85 F. November, December, January and February are the temperate months when the mean average temperature is about 79 F. Residents divide the year into two seasons, wet and dry. Rains are heaviest in July, August and September. The highest yearly rain average in the islands will sometimes reach 125 inches, although Manila's rainfall is usually 50 inches or so below that.

Air conditioning engineers who have designed foreign installations are acquainted with the fact that high dry bulb temperatures are rare in the tropics, although certain parts of India and Africa may be considered as exceptions to this rule. Generally speaking, however, a dry bulb temperature of 90 to 95 F is maximum, with 90 F the most common. However, the wet bulb temperatures at such times may run around 80 to 85 F, a condition that makes the heat particularly oppressive and difficult to withstand. Naturally, lowering of the dew-point temperature by condensing out a portion of the moisture content of the air can bring welcome relief.

The air conditioning engineer is also certain to bump his head against the problem of tropical architecture. The majority of houses and buildings in Manila, for example, were built with the idea of allowing the maximum

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amount of air circulation. They are extremely lightweight structures with many large windows, open passage-ways, and latticed partitions. Many of the windows are made up of countless little pieces of "shell glass" which are not transparent, and for this reason, the windows are arranged to slide or swing out in order to provide vision for the occupants of the room. Windows and doors seldom fit closely, as there is never any regard for the infiltration of outside air in a climate where a temperature of 65 F is a real "cold wave." Naturally, all of this complicates the problem of confining the air supply to the spaces to be conditioned, and the general design and construction of many of these buildings is such that the leakage of heat from the exterior is much higher than we have to contend with in the temperate zone where all buildings have been designed to resist cold weather.

This light construction has been no doubt influenced by native demands for cheaper construction which would still provide the greatest amount of comfort. In Manila, the visitor also notices a decided Oriental influence in many of the buildings. Bamboo floors, false ceilings of woven grass mats, sliding partitions, large windows and high peaked roofs are all adopted Oriental aids to ventilation. The heat of the day penetrates easily, of course, but it is soon dissipated after the sun goes down. Buildings of this type, which delight the eye of the casual visitor with their picturesque appearance, are something of a head-ache to the air conditioning engineer, and because they are constructed on a principle directly opposed to air conditioning standards, they must be practically rebuilt to achieve any degree of air conditioning success.

On the other hand, a number of buildings and most of the better class homes are constructed with heavy masonry walls. They absorb the extreme heat of the sun during the day and give it up during the comparatively cool nights. The factor of time lag is such on a thick masonry wall that the sun's heat never quite penetrates the interior before the day is over, and it turns back in its course to be absorbed by the cooler night air. As a result, the interiors of these buildings remain somewhere between the averages of the dry bulb and wet bulb temperatures which prevail throughout the 24 hours.

These buildings with more of an Occidental design reflect the construction work of the Spaniards and are identical with many buildings of the same type to be found in Latin-American countries. More modern office buildings, hotels, apartments, hospitals, etc., are following more closely the typical American standards of design and construction and are, of course, easily adaptable for air conditioning.

Despite the problems to be conquered, air conditioning has an excellent future in tropical countries. Naturally, a great deal of pioneering and educational work will be necessary to develop this market, but Manila, as an example, shows an amazingly high acceptance for air conditioning. The public is aware of it and highly appreciative of it. Theaters were the first to introduce comfort cooling, but air conditioned offices, hotels and stores are fast becoming popular. While there, I had occasion to supervise the installation of an air conditioning system in the new Southern Investments and Management Corp. offices. Executives of the new firm did not question the advisability of including air conditioning in their new quarters; it was, they felt, a necessary part of the service they offered their patrons, and this concurs with the growing opinion most tropical business men have of air conditioning.

[*Concluded from page* 560]

corresponding C_x and F_x points; draw the water lines from these points and find so the corresponding B_x and G_x points. These two must be on the same horizontal line t_m. Modify until this requirement is fulfilled.

Two Stage Split Spray Air Washer

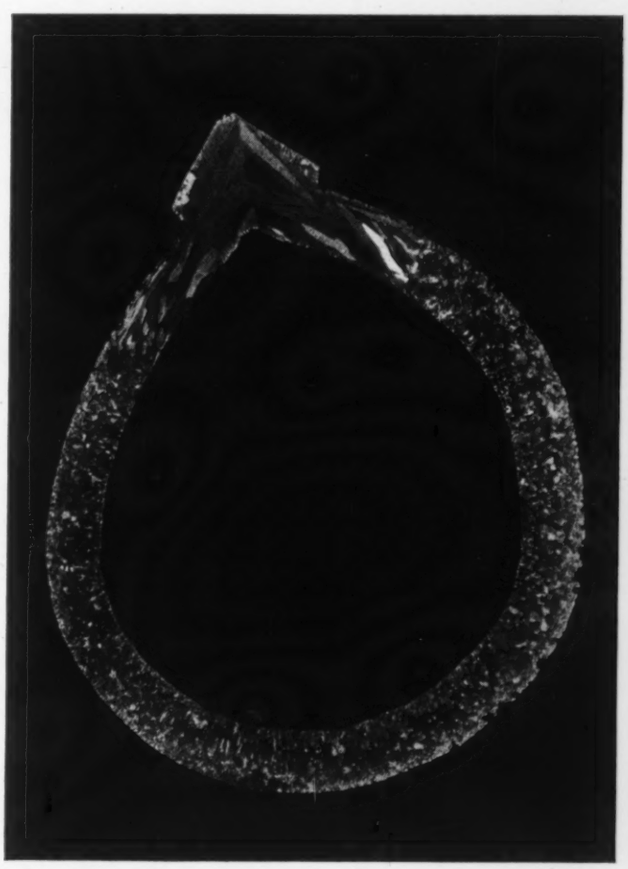
This type of air washer is similar to the previous one with the exception that both banks of nozzles receive the spray water with the same initial temperature t_1. Fig. 5 illustrates the conditions. The essential points and lines are designated as for the previous type. However, since t_1 is the initial water temperature for both stages, points B and F are located on the same t_1 temperature line. The water rate W_1 may be different for each stage: $tg\alpha = W_{11}$ and $tg\beta = W'_{12}$. Accordingly the slope of the two water lines will be different, *i. e.* they are not necessarily parallel, although the usual way is to have W_1 the same for both stages, since this arrangement gives the best results. In this case $\alpha = \beta$.

[*Concluded from page* 561]

had reached 250 lb. per sq in. Furthermore, failure did not occur at the seams, which proved to be a source of strength rather than of weakness.

The assistance of F. G. Skinner, Esq., of The Science Museum, London, and of V. A. Yardley, Esq., of the Chemical Research Laboratory, Teddington, Middlesex, England, is gratefully acknowledged.

Fig. 3—Macro-etched section of pipe shown at extreme left of Fig. 2



E. Vernon Hill's* Personal Comments

Heat and Cold

TO the engineer of long years experience most of the terms we use have a clear and definite meaning, even though they lack descriptive value and accuracy. But to the novice and the student, our nomenclature dealing with engineering subjects is confusing and often misleading.

We speak, for example, of the manifestations of heat as if there were many kinds—latent heat, sensible heat, radiant heat, convected heat, conducted heat, etc. This conveys to the mind of the uninitiated the thought that there are actually different kinds of heat, and we realize when talking with students that they actually believe this to be true. Of course it is not so. Heat is energy. It is the evidence of molecular motion and there is only one kind. A man may look different in front view, profile, or from behind, but we do not have different names for him because he presents a changed appearance from different angles.

Heat may be lost from a warm body to one of lower temperature, or to the air, by convection currents, by conduction, or by radiation, but for the benefit of those who are not familiar with these terms we should be a little more careful to state the facts accurately and not give the impression that convected heat, conducted heat, and radiant heat are different forms of energy.

We speak also of latent heat and sensible heat as two very dissimilar manifestations, and it is frequently stated that sensible heat is heat that is perceived or felt by living organisms while latent heat is not. This conclusion does not bear close analysis. We assume, without sound reason, that because so-called latent heat in the air does not affect a dry bulb thermometer that it does not affect the living organism. As a matter of fact this is not true. Our sensations of warmth are definitely affected by latent heat in the air as well as by the heat we call sensible. The research work carried out by the investigators in the *A.S.H.V.E.* laboratories in Pittsburgh on the effective temperature bears out this assertion.

The latent heat in the water vapor of the air is manifested as an accelerated molecular motion. Sensible heat produces the same molecular effect. It is logical to assume that if one affects the nerves of the skin the other will have the same effect.

When heat is added to water vapor, increasing thereby its temperature above the temperature of saturation, we speak of it as superheat; but we do not speak of the sensible heat in the air as superheat, although in the latter case the term is as truly applicable as in the former. Liquid air absorbs heat, which we may term latent, to change it from a liquid to a vapor, and any heat the air contains above its saturation temperature can properly be termed superheat.

But superheat is the stumbling block of every student. If we would call it sensible heat of the vapor the meaning would be clear and the term superheat could be dispensed with.

The term convected heat—or better, the heat loss by convection—is of questionable value. The transfer of heat from a warm surface to the air in contact with it is a process of conduction, modified, of course, by the velocity of the air passing over the heated surface. But we do not use this confusing term when discussing the transfer of heat from a warm to a cold liquid separated by a metal partition, as in a condenser for example, although the same or similar conditions prevail. And why we should confuse our minds with the term convection, in considering the transfer of heat from a warm surface to the air, is hard to explain or excuse.

Radiant heat is frequently spoken of as a form of energy quite distinct and dissimilar from other heat manifestations. Probably this is because its method of transmission through a vacuum, or even through the air, is not well understood; and yet all the heat from every source that can be convected, conducted, transferred, or generated, on the earth's surface or from its depths, is or was originally all radiant heat from the sun.

And then we talk of cold as if it were a separate entity. We say air is warm or cold, but the expression has little meaning except as a relationship with some other condition. Standing in front of a unit cooler on a hot day we say the air leaving the machine at a temperature of 50 degrees is cold, but a breeze blowing from the south in the winter time at the same temperature we would term exceedingly warm. We say our coffee is cold when in fact it is well above 100 degrees in temperature, and we say a glass of drinking water is warm when in reality it is quite cold compared with the coffee.

The informed man, of course, realizes that the term cold is a misnomer and if you ask him for an explanation he will tell you that there really is no such thing as cold—"cold is the absence of heat." But this explanation is obviously untrue. The absence of heat is a theoretical condition where molecular motion ceases. Cold as we term it is only a different degree of heat intensity and a substance is cold or warm only in relation to some other substance of different temperature by comparison.

A lady I am supposed to know quite intimately enjoys sleeping on our unheated enclosed sleeping porch when the weather permits. During the spring months she complained one morning of not sleeping well on account of the cold, even with ample blankets. She explained to me at the breakfast table that the cold "comes up through the floor" and makes her uncomfortable even though adequately covered. I chided her for her unscientific statement. "Cold does not come up through the floor," I said. "In fact, there is no such thing as cold. Cold is merely the absence of heat."

She looked at me with obvious skepticism, but said no more about it. The next night was also rather cold and at the breakfast table, while I was on the second trip around my grapefruit, I asked her how she rested during the night. "Oh, fairly well," she said, "but the 'absence of heat' is still coming up through the floor."

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Selecting the Right Size Heating Boiler†

By Sabin Crocker*

The Warming-Up Problem Explained

If heating systems were kept warm at all times from fall to spring by a type of steady firing which adjusted itself perfectly to the weather, there would be no occasion for a warming-up allowance. This condition is seldom, if ever, realized in building heating for a number of reasons, some of which follow:

(1) Heat is not required on mild days in spring and fall, but must be brought up quickly on short notice due to a change in weather—say on a chilly morning following the end of a mild period after the building has had a chance to cool over night. (2) With hand firing the system and building may cool before the fire is replenished and need to be brought up to temperature again quickly. (3) With automatic firing the building may be allowed to cool 10 to 20 F at night below the day-time temperature and then be brought up again quickly in the morning. (4) Buildings such as churches and auditoriums which are heated on occasion must be brought up to the desired temperature within a reasonable length of time.

A further reason for some warming-up allowance in any system is the extra load represented by a fall in air temperature inside the building. If heat is cut off for any reason and the air temperature in the building falls below the design temperature, an added load is thrown on the heating system due to the increased heat emission in cooler air. According to Table 2 (on p. 3) a fall in room temperature of 10 F increases the heat emission about 9 per cent, so that a fall in room temperature of 10 F puts a 9 per cent extra starting tax on the system.

When occasion warrants, it is possible to substitute rational engineering methods in place of empirical rules and compute the size boiler required to produce a desired result. Some assumption regarding a satisfactory time to bring this entire system to the operating temperature is required as a basis. What constitutes a satisfactory time varies with different applications, but probably 1 hr is about the average time to expect for a steam system from a "stone cold" start, represented for instance by room temperature and boiler temperature both at 50 F with the fire out. The 1 hr would not include time required for kindling a coal fire or for bringing the room temperature up to, say, 70 F after raising the system to operating temperature.

Computing the Time Required

Computation of the total time required involves the following steps in the case of a steam system:

(*a*) Assume what seems to be the right size boiler using one of the empirical rules, say that of the *A. S. H. V. E.*

(*b*) Compute the time required to bring the boiler and its water content from 50 F up to the operating temperature. This involves computing the total amount of sensible heat needed to raise the temperature of boiler

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†Summarized and revised from a series of three articles published in Heating, Piping and Air Conditioning, February-March-April, 1932.

obtained by counting their center line measurements as so much pipe, since welded fittings commonly are of the same thickness as the pipe with which they are used. Some additional allowance is required for valves and such flanges and flanged fittings as are used. This allowance probably will not run more than 5 to 10 per cent of the weight of pipe and welded fittings.

Examples of the method of computing the warming-up period may be found in the original article.

Table 9—Approximate Weight in Lb of 125 Lb Standard Cast Iron Flanged Fittings

Nominal Size, In.	90° Ells	45° Ells	L. R. Ells	Base Ells	Tees	Crosses	Laterals	Scd. Flgs.	Blind Flgs.	Flgd. Unions
½	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
¾	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
1	.....	.....	.....	.....	.....	.....	.....	2	2	3
1¼	.....	.....	.....	.....	.....	.....	.....	3	3	4
1½	.....	.....	.....	.....	.....	.....	.....	4	4	5
2	18	15	.....	.....	26	34	.....	5	5	6
2½	22	20	.....	.....	34	43	.....	7	7	9
3	30	27	.....	.....	45	58	.....	8½	9	12
3½	37	33	.....	.....	55	74	.....	11	13	15
4	41	38	60	70	67	89	90	12	17	17
4½	46	43	70	.....	75	100	100	13	18	21
5	63	53	80	100	90	121	120	15	21	25
6	75	68	100	125	115	152	140	18	27	34
7	100	90	150	150	150	200	190	25	37	48
8	120	100	185	190	170	236	230	32	45	57
9	150	130	225	230	220	305	310	35	56	65
10	205	160	275	270	285	400	325	44	67	84
12	285	230	410	370	430	570	600	65	99	116
14	390	300	600	540	550	750	750	95	133	.....
15	440	330	675	.....	660	800	925	100	150	.....
16	525	400	875	700	760	1000	1000	105	175	.....
18	.....	.....	.....	.....	.....	.....	.....	125	215	.....
20	.....	.....	.....	.....	.....	.....	.....	175	281	.....
22	.....	.....	.....	.....	.....	.....	.....	195	347	.....
24	.....	.....	.....	.....	.....	.....	.....	230	422	.....

Table 10—Approximate Weight in Lb per Sq Ft of Rated Surface for Cast Iron Radiators[a]

Type of Radiator	Cast Iron	Water Content
Column	7.50	1.00
Tube	5.50	1.00
Light weight	4.125	0.75
Wall	7.00	1.40

[a]The specific heats in Btu per lb per F are as follows: water = 1.00; cast iron = 0.13.

Here is the fourth installment of a new series of "summary sheets," on this and the preceding page. The purpose of these sheets is to make available for quick reference for everyday use information needed in heating, piping and air conditioning work. . . . For complete information, refer to the original article from which these data were obtained, as the purpose of these summaries is not to take the place of the article, but rather to put the essentials into convenient reference form. . . . The Editors will appreciate your suggesting what type of material you'd like presented in this way. . . . Address HEATING, PIPING AND AIR CONDITIONING, 6 North Michigan Avenue, Chicago, Ill.

metal and water content, which amount then is divided by the rate of heat absorption through the boiler surface to obtain the length of time required. In case the boiler is well insulated, heat loss from the boiler exterior can be neglected, otherwise some allowance should be made for this loss.

(*c*) Compute the weight of metal in covered piping. This part of the system will get warm next after the boiler and its time interval should be computed as a separate item, assuming that the full output of the boiler will be available for this purpose during the time interval involved. Radiation loss from this piping will be small, but can be taken into account by assuming that the normal covered pipe loss of 50 F air temperature exists during this period, which compensates in part for heat absorbed in raising the temperature of the covering.

(*d*) Compute the weight of metal in bare piping and radiators, and from this the amount of heat required to bring them from 50 F to operating temperature. The full output of the boiler will not be available for this purpose since the covered pipe tax must be carried as well as picking up the heat emission load of radiators and bare piping as they warm up in rooms at temperature lower than normal. All these factors can be taken into account by an approximate heat balance method, or by the use of formulas or curves relating the factors.

Item *a* above already has been explained. Item *b* can be computed from data obtained from the boiler manufacturer, or in case this is not convenient the typical values of Table 6 can be used as approximations for the types of boilers indicated.

Items *c* and *d* can be computed from Tables 7-10 giving the weights of pipe, screwed and flanged fittings and radiators respectively.

Weight of Pipe and Fittings

Computing the weight of pipe in a given heating system is a comparatively easy job, while computing the weight of the variety and size of fittings used is somewhat tedious. For this reason it may be convenient to express the weight of fittings as a percentage of the pipe weight. Analysis of typical steam and water jobs containing a mixture of flanged and screwed fittings indicates that the fittings and valves weigh approximately half as much as the pipe. Where screwed fittings are used exclusively, their weight may run 20 to 25 per cent that of the pipe, while with flanged fittings the weight may run to 75 per cent that of the pipe. In flanged boiler

Table 6—Typical Weights of Boilers and Water Content in Lb per Sq Ft of Heating Surface[a]

Boiler Heating Surface Sq Ft	Metal Weight		Water Weight			
			Steam Boiler		Water Boiler	
	Cast Iron Sectional	Steel Portable Fire-box Type	Cast Iron Sectional	Steel Portable Fire-box Type	Cast Iron Sectional	Steel Portable Fire-box Type
15–25	55–65	40–45	8	.....	11	.....
25–50	45–55	35–40	7–8	12–14	10–11	16–19
50–100	40–50	25–35	6–8	12–14	9–11	14–19
100–500	35–40	15–25	6–7	9–14	9–11	14–19
500–1000	30–35	12–22	6	9–10	10	13–14
1000–3000	30–35	10–17	.....	7–10	.....	10–14

[a]The specific heats in Btu per lb per F are as follows: water = 1.00; cast iron = 0.13; steel = 0.117.

Table 7—Weight per Lineal Foot of Standard Weight and O. D. Steel and Wrought Iron Pipe with Plain Ends[a]

Nominal Size, In.	Thickness, In.	Metal Weight per Ft Lb	Water Weight per Ft Lb	Nominal Size, In.	Thickness, In.	Metal Weight per Ft Lb	Water Weight per Ft Lb
½	0.109*	0.850	0.13	5	0.258*	14.617	8.67
¾	0.113*	1.130	0.23	6	0.280*	18.974	12.51
1	0.133*	1.678	0.37	8	0.322*	28.554	21.68
1¼	0.140*	2.272	0.65	10	0.365*	40.483	34.19
1½	0.145*	2.717	0.88	12	0.375	49.562	48.94
2	0.154*	3.652	1.45	14 O.D.	0.375	54.568	59.78
2½	0.203*	5.793	2.07	16 O.D.	0.375	62.579	79.18
3	0.216*	7.575	3.19	18 O.D.	0.409	76.840	100.5
3½	0.226*	9.109	4.29	20 O.D.	0.409	85.577	124.8
4	0.237*	10.79	5.50	24 O.D.	0.4375	110.10	181.0

[a]The specific heats in Btu per lb per F are as follows: steel = 0.117; wrought iron = 0.114; cast iron = 0.13; water = 1.00. Density of water is 62.4 lb per cu ft.

The weight of couplings, flanges, fittings and valves runs about 50% that of the pipe metal for average jobs with a mixture of flanged and screwed joints. In welded jobs, count welding fittings as pipe and add about 10% for the weight of valves, flanges and flanged fittings.

*These wall thicknesses correspond to Schedule 40 pipe of the standard for wrought iron and wrought steel pipe, *A.S.A.* B36.10.

Table 8—Approximate Weight in Lb of 125 Lb Standard Cast Iron Screwed Fittings

Nominal Pipe Size, In.	90° Ell	45° Ell	Tee	Cross	Y Bend	Union[a]	Reducer 1 Size	Bushing 1 Size	Cap	Plug, Square Head Regular	Plug, Square Head Solid
¼	0.15	0.15	0.25	0.40	....	0.20	.....	0.03	.....	0.03	.....
⅜	0.25	0.20	0.35	0.40	0.60	0.30	.....	0.04	.....	0.06	.....
½	0.40	0.35	0.55	0.70	0.60	0.40	.....	0.06	.....	0.08	0.10
¾	0.65	0.55	0.85	1.00	1.00	0.60	.....	0.12	.....	0.15	0.20
1	1.00	0.85	1.50	1.75	1.75	1.00	.....	0.20	.....	0.20	0.30
1¼	1.50	1.25	2.00	2.50	2.50	1.25	1.25	0.35	1.00	0.35	0.55
1½	2.00	2.00	2.75	3.50	3.50	1.75	1.75	0.45	1.50	0.50	0.75
2	3.00	2.75	4.25	5.25	5.25	2.75	2.25	0.70	2.00	0.80	1.25
2½	4.75	4.75	6.50	8.00	8.50	4.50	3.00	1.25	2.75	1.25	2.00
3	7.25	6.50	9.50	11.00	12.00	6.00	4.75	1.75	4.50	2.00	3.00
3½	9.50	8.50	14.00	16.00	17.00	9.75	6.25	2.50	5.75	2.75	4.00
4	12.00	11.00	18.00	21.00	24.00	13.00	8.00	2.75	6.25	3.25	5.50
5	22.00	18.00	28.00	32.00	41.00	26.00	13.00	4.25	11.00	5.25	10.00
6	30.00	27.00	40.00	45.00	57.00	40.00	19.00	7.25	16.00	7.50	14.00
8	62.00	44.00	79.00	88.00	121.0	50.0	31.00	11.00	31.00	16.00	27.00
10	116.0	88.00	154.0	170.0	252.0	80.0	56.00	21.00	50.00	26.00	51.00
12	178.0	136.00	230.0	271.0	384.0	120.0	83.00	31.00	80.00	41.00	75.00

[a]Std. weight malleable iron ground joint type up to and including 4 in.; cast iron flanged union with bolts 5 in. and larger.

outlet headers and the like, the weight of fittings may be several times the weight of pipe. Where direct cast iron radiation is used, the combined weight of fittings, pipe and valves usually will be from 2 to 3 lb per sq ft of standing radiation, the higher value applying to water systems or where flanged fittings predominate.

With welded jobs the approximate weight of all welded fittings can be

Heating · Piping
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AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

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Cooling Load Analysis of a Bank Building

By John James* (*MEMBER*), New York, N. Y.

HOW do design engineers appraise the cooling load factors of solar radiation, uncontrolled infiltration and controlled ventilation? What is customary practice in determining cooling load requirements for the selection of equipment? When are safety factors used and how much? Why are bidders' proposals so far apart for air conditioning a given enclosure? How will future research investigations tend to eliminate many of the unknown variables? Answers to these important questions are discussed in this paper, based on a survey conducted in 27 of the United States and several provinces of Canada and the results are particularly significant.

Load factors for heating are fairly well established, but scientific data are still lacking for accurately estimating the several components of the cooling load, which accounts in part for the lack of uniformity in cost estimates and system designs. Practically every engineer and manufacturing organization has devised forms for calculating a cooling load, and it is interesting to note that many of them are based on Society research data. Some forms of computation are very comprehensive taking into consideration all of the various components of the load as they are affected by the particular building design. Other forms are based on empirical values which are assumed to be typical or average figures that can be used on all types of building construction with sufficient accuracy.

Scope of Survey

A method of cooling load computation is outlined in Chapter 8 of THE A.S.H.V.E. GUIDE, 1937, which is based on numerous investigations conducted at the Research Laboratory and cooperative institutions. A typical one story banking space problem shown in Fig. 1 was forwarded to members of the Society with a request that a sample cooling load calculation be prepared in accordance with their usual practice. It was thought that a survey of this kind would indicate whether THE GUIDE method was actually being used in the field and, if not, what other procedures were in general practice.

In response to an invitation extended to engineers in every state and province of Canada, 47 calculations were returned representing most of the principal territories requiring summer air conditioning. These replies were submitted from technical men of the following professional status: 21 consulting engineers, 11 manufacturer's engineering representatives, 8 utility engineers, 4 contractors and 3 professors.

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It was the intent of the survey to limit the study primarily to factors affecting the solar radiation transmitted, uncontrolled infiltration air, and controlled ventilation air. Hence, it was found desirable to reduce the calculations to a minimum of variables with general design conditions as follows:

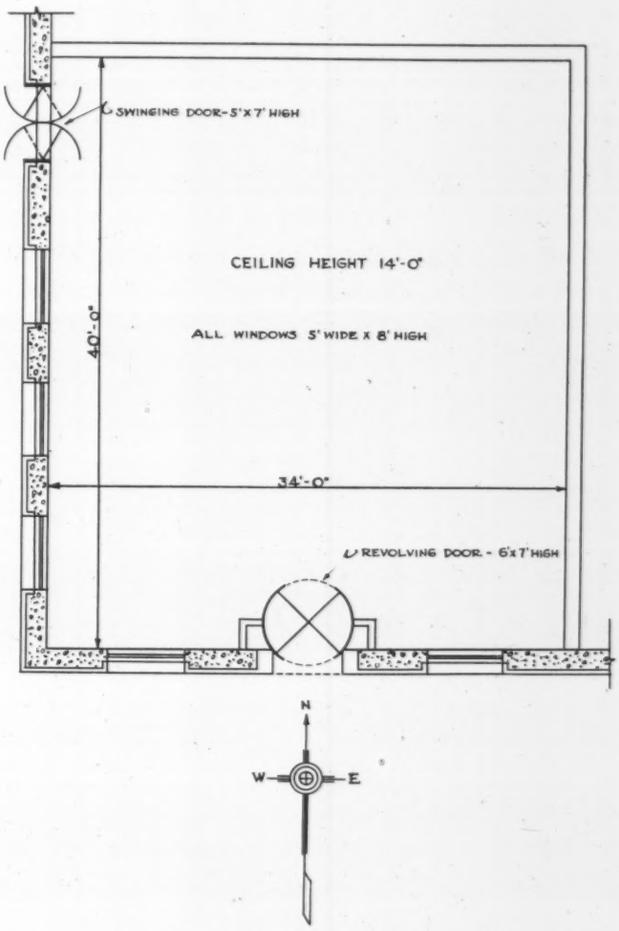

Fig. 1—Plan diagram of banking space

Building Construction:

Outside Wall, $U = 0.37$ for 4 in. red brick veneer, backed with 16 in. concrete and ½ in. plaster.

North and East Inside Walls—no heat gain.

Flooring—no heat gain.

Roof, $U = 0.26$, for 2 in. concrete roof deck with black finish roofing material, ½ in. rigid insulation and suspended plaster ceiling.

Windows, $U = 1.13$ for single glazed double-hung metal non-weatherstripped.

People load, 26 individuals.

Light load, 4,500 watts.

Assume no shading effects from adjacent structures in calculating sun effect.

In order to unify the tabulation of the results, it was requested that the calculation sheet show individual determinations of the sensible, latent and total cooling loads. So that values in THE GUIDE could be compared, it was suggested that the design inside and outside dry- and wet-bulb temperatures be given. Other significant items requested in the statement of conditions were a complete explanation of the assumptions used in determining the solar radiation effects through windows, vertical walls and roof. Also, the stated amount of outside air introduced for ventilation was to be given with values for the air infiltration through the windows, swinging doors and revolving doors.

Sample Load Calculation

Using the stated conditions for the problem and assuming certain design conditions, a sample calculation of the cooling load is indicated in Table 1 based on the method outlined in THE GUIDE 1937. The computation is divided into the various components of both the sensible and latent heat gains. Transmission and sun effect are determined from the degrees which are recommended to be added to the outside dry-bulb temperature for different types of surfaces. THE GUIDE indicates methods for the determination of infiltration based on the amount of air per lineal foot of window crack and also on the number of air changes taking place depending on the kind of room or building. Checking the total amounts of air entering the enclosure by these two methods shows that the results in each case are less than the specified amount recommended in Chapter 3 of THE GUIDE for the quantity of outside air introduced for ventilation to take care of the human occupancy requirements. Therefore, it is assumed in this case that the ventilation load is based on the controlled outside air, thus resulting in a positive pressure within the enclosure and causing exfiltration through the window and door openings.

Design Outside and Inside Conditions

A primary consideration in a cooling load determination is the selection of the correct outside and inside dry- and wet-bulb temperatures. Frequently, engineers in the same locality have divergent views in respect to the proper temperatures to assume. Values given in THE GUIDE for heating load calculations are merely a compilation of climatic conditions obtained from government Weather Bureau records and even though the lowest temperatures are stated, engineers rarely base their design on these values. Such temperatures are usually of short duration and are rarely repeated in successive years. When designing a heating system it is customary practice to assume an outside temperature of 15 F above the lowest recorded temperature.

Likewise, in the design of a cooling system it is important to exclude these maximum outside temperatures

Table 1—Sample Cooling Load Calculation Based on THE GUIDE Method

DESIGN CONDITIONS

Outside 100 F dry-bulb 75 F wet-bulb 87 grains per pound dry air
Inside 80 F dry-bulb 67 F wet-bulb 77 grains per pound dry air

SENSIBLE HEAT GAIN		BTU PER HOUR
Transmission and Sun Effect		
South Wall	$= 354 \times 0.37 \times [(100 + 10) - 80] =$	3,930
West Wall	$= 405 \times 0.37 \times [(100 + 20) - 80] =$	6,000
North Wall, no heat gain		
East Wall, no heat gain		
South Glass	$= 122 \times 90$ (2 p. m.) $\times 0.97 =$	10,680
West Glass	$= 155 \times 140$ (2 p. m.) $\times 0.97 =$	21,000
Roof	$= 1,360 \times 0.26 \times [(100 + 45) - 80] =$	23,000
People Load	$= 26 \times 220 =$	5,720
Lighting Load	$= 4,500 \times 3.413 =$	15,300
Ventilation, assume 15 cfm per person outside air introduced with exfiltration due to positive fan pressure of system.		
V_s	$= (15 \times 26) \times 60 \times 0.24 \times 0.0735 \times (100 - 80) =$	8,250
	Total Sensible Heat Gain =	93,880
LATENT HEAT GAIN		
People Load	$= 26 \times 180 =$	4,680
Ventilation V_1	$= (15 \times 26) \times 60 \times 0.0735 \times \frac{87-77}{7,000} \times 1,060 =$	2,600
	Total Latent Heat Gain =	7,280
TOTAL COOLING LOAD		101,160
Tons of Refrigeration	$101,160 \div 12,000 = 8.42$	

Table 2—Comparative Design Outside and Inside Conditions for Cooling Load Determinations

No.	State	City	THE GUIDE 1937 TABLE 1 CHAP. 8 OUTSIDE TEMP. DEG F D. B.	THE GUIDE 1937 TABLE 1 CHAP. 8 OUTSIDE TEMP. DEG F W. B.	ANALYSIS OUTSIDE TEMP. DEG F D. B.	ANALYSIS OUTSIDE TEMP. DEG F W. B.	ANALYSIS INSIDE TEMP DEG F D. B.	ANALYSIS INSIDE TEMP DEG F W. B.	EFF. T EMP.	R. H. %
1	Ala.	Birmingham	93	77	95	78	80	67.0	74.0	50
2	Calif.	Los Angeles	88	70	95	74	75	62.5	70.0	50
3		Los Angeles	88	70	90	70	77	64.5	71.7	50
4		Fresno			100	69	80	65.5	73.4	45
5		Oakland (San Francisco)	85	68	85	67	75	63.0	70.1	50
6	D. C.	Washington	95	78	95	78	80	65.0	73.2	44
7	Fla.	Miami Beach (Tampa)	94	79	91	79	79	67.0	73.6	53
8	Ill.	Chicago	95	75	95	75	80	67.0	74.0	50
9		Chicago	95	75	95	75	80	67.0	74.0	50
10		Chicago	95	75	95	75	80	67.0	74.0	50
11		Urbana (Peoria)	91	75	91	75	80	67.0	74.0	50
12	Ind.	Indianapolis	90	73	95	77	78	65.0	72.3	50
13	Iowa	Des Moines	92	74	92	74	78	64.5	72.1	49
14	Ky.	Lexington (Louisville)	94	75	94	75	82	67.0	75.0	45
15	La.	New Orleans	94	79	95	79	80	67.0	74.0	50
16		New Orleans	94	79	95	79	80	67.0	74.0	50
17	Md.	Baltimore	93	76	95	78	80	67.0	74.0	50
18		Baltimore	93	76	95	78	80	67.0	74.0	50
19	Mass.	Boston	88	73	90	75	78	66.0	72.7	54
20	Mich.	Detroit	93	73	95	75	80	67.0	74.0	50
21		Grand Rapids			95	75	80	67.0	74.0	50
22		Kalamazoo			95	78	80	67.0	74.0	50
23	Mo.	Kansas City	92	75	100	75	80	67.0	74.0	50
24		Kansas City	92	75	100	78	80	67.0	74.0	50
25	Mont.	Billings (Helena)	87	63	100	71	82	67.0	75.0	45
26	Nebr.	Omaha (Lincoln)	93	74	95	78	80	67.0	74.0	50
27	N. J.	Newark (Trenton)	95	76	95	79	80	65.0	73.2	44
28	N. Y.	Buffalo	83	72	95	75	80	67.0	74.0	50
29		New York	95	75	95	75	80	67.0	74.0	50
30	N. D.	Bismark (Grand Forks)	88	69	86	68	77	64.5	71.7	50
31	Ohio	Cincinnati	95	78	95	78	80	67.0	74.0	50
32		Cincinnati	95	78	95	78	80	65.0	73.2	44
33		Cleveland	95	75	95	75	80	67.0	74.0	50
34		Columbus			95	75	80	66.0	73.6	47
35		Columbus			95	77	80	65.0	73.2	44
36		Dayton			95	78	80	67.0	74.0	50
37	Okla.	Oklahoma City	96	76	96	76	81	65.9	74.0	44
38	Oreg.	Portland	83	65	85	65	75	65.0	70.8	59
39	Pa.	Pittsburgh	91	73	90	73	78	64.0	72.0	46
40		Pittsburgh	91	73	95	75	80	67.0	74.0	50
41	Tenn.	Memphis	93	77	95	78	80	67.0	74.0	50
42	Texas.	Ft. Worth (Dallas)	99	76	100	78	83	67.0	75.5	43
43	Vt.	Burlington	85	71	90	73	80	68.0	74.5	55
44	Wash.	Seattle	83	61	84	63	70	60.0	66.4	56
45	Wis.	Milwaukee	93	74	95	75	80	67.0	74.0	50
46	Canada	Winnipeg			90	70	76	66.0	71.8	59
47	Canada	Toronto			90	72	75	66.0	71.1	61

Table 3 — Recommended Indoor Air Conditions in Summer Corresponding to Outdoor Temperatures

Outside Design Dry-Bulb Temp. Deg F	Inside Design Effective Temperatures	
	The Guide[a] 1937 Table 2, Chap. 3	Table in Proposed Code[b] of Min. Requirements for Comfort Air Cond.
105	.....	75.5
100	.....	75.0
95	73.0	74.0
90	72.0	73.0
85	71.0	72.0
80	70.0	71.0
75	69.0	.....
70	68.0	.....

[a] Applicable to exposures less than 3 hours.
[b] For occupancies in excess of 40 min and for sedentary occupations, between relative humidity limits of 30 and 70 per cent.

which occur for short periods of duration. Values given in Table 1, page 158, of The Guide 1937 are design temperatures which are based on a study of hourly temperatures in New York City from which factors were derived and applied to the average maximum dry- and wet-bulb temperatures for other cities. In some cases revisions have had to be made in the original table based upon the recommendations of committees appointed to study weather conditions by the local Society Chapters.

It should be recognized that the type of building occupancy governs in many respects the selection of proper outside design temperatures. The values given in The Guide are for an installation operating approximately 10 light hours a day or for a cooling season of 1,200 hours in June, July, August and September for an average year. If the system is being designed for an industrial plant operating 24 hours a day or for a theater operating from 11:00 a. m. to 11:00 p. m., it is obvious that a study of the hourly temperatures would show different results because of the cooler night air conditions.

Outside Design Temperatures

Based on the results obtained from this survey, values are given in Table 2 showing the design outside and inside temperature conditions for various localities. The first two columns of the tabulation show figures corresponding to the values given in The Guide and the latter columns refer to the results reported in an analysis of the survey. In many instances the temperatures obtained from the survey check closely with The Guide values and the extreme variations occur only in the following cities: Los Angeles, Indianapolis, Baltimore, Boston, Detroit, Kansas City, Billings, Omaha, Newark, Buffalo, Pittsburgh, Ft. Worth, Burlington and Seattle.

A Joint Committee[1] of the A.S.H.V.E. and *American Society of Refrigerating Engineers* recently considered a re-study of design outside temperatures for heating and cooling load estimates and as a result a Society Research Technical Advisory Committee was appointed to develop a formula for appraising the proper design conditions from Weather Bureau records. Upon completion of a satisfactory formula or standard to use for this investigation, it is expected that the Research Laboratory will arrange for a comprehensive survey of all Weather Bureau records in the principal cities of this country and Canada.

[1] A. S. H. V. E.-*A. S. R. E.* Committee on National Standards for Air Conditioning Applications.

Inside Design Temperatures

The problem of selecting the proper indoor air conditions for a given enclosure is one which is open to considerable discussion and debate. Recent research seems to indicate that effective temperature is important only from a standpoint of comfort and that the dry- and wet-bulb temperatures may vary over a considerable range if the proper effective temperature is maintained within limits of the summer comfort zone. Several references are given in Chapter 3 of The Guide outlining the selection of the proper effective temperature to use in the design of a system for buildings of various occupancies. For several years data have been included in The Guide, shown in the second column of Table 3 of this paper, on recommended indoor air conditions in summer corresponding to various outdoor temperatures. The Guide values have been subject to much discussion but up to the present time no satisfactory substitute has been offered. Many uncontrollable variables enter into this problem and it is expected that work now being conducted at the Research Laboratory and other interested institutions may throw considerable light on this complex problem.

In the last column of Table 3 are values suggested in a proposed Code now available for discussion as submitted by the Joint Committee.[2] It states on page 72 of The Guide 1937, for banks and stores where the period of occupancy is short, that the contrast between outdoor and indoor air conditions becomes the deciding factor in regard to the maintenance of proper temperature and humidity. General experience favors a schedule of design for banks about 1 deg effective temperature higher than that shown in column 2 of Table 3. It will be noted that if this recommendation is followed the values proposed for the Code would check quite closely with the figures now given in The Guide.

Table 4 is arranged so as to give the number of inside design effective temperatures for various cities corresponding to the outside design dry-bulb temperatures as taken from Table 2. With only outside design dry-bulb temperatures used as a basis of comparison, it will be noted that the survey indicated a close check with values in Table 3 for conditions corresponding to 95 F outside and 74 deg inside effective temperature.

Climatic Conditions in Other Localities

Incomplete data were supplied by a few engineers to comprise a complete analysis for the banking problem but the information forwarded regarding climatic conditions is of particular significance in connection with this discussion.

Philippine Islands

Complete temperature data seem to be lacking in Manila, but general observations show wet-bulb temperatures of from 77 to 79 F with maximum dry-bulb temperatures varying between 80 and 97 F. Outside design conditions for this locality are generally assumed to be based on 93 F dry-bulb and 79 F wet-bulb for cooling load determinations. Desirable indoor conditions are selected on a basis determined by taking two-thirds of the difference between the outdoor tempera-

[2] Loc. Cit. See Note 1.

Table 4—Design Inside Temperatures Corresponding to Outside Temperatures

Outside Design Dry-Bulb Temp. Deg F	Number Inside Design Effective Temperatures Corresponding to Outside Design Dry-Bulb Temperatures																
	75.5	75.0	74.5	74.0	73.6	73.4	73.2	72.7	72.3	72.1	72.0	71.7	71.1	70.8	70.1	70.0	66.4
100	1	1		2		1											
96				1													
95				20	1		5		1							1	
94		1															
92										1							
91				1	1												
90			1					1			1	1	1				
86												1					
85														1	1		
84																	1

ture and 75 F dry-bulb with relative humidity limits of 40 to 60 per cent. Recommended inside temperatures corresponding to outside conditions are shown in Table 5.

Denver, Colorado

Engineers familiar with sea level atmospheric conditions rarely appreciate the design differences which exist at high altitudes when applied to the selection of air conditioning systems. Information obtained from Denver emphasizes that the mountain states of that region have a wet-bulb temperature not over 60 F practically 95 per cent of the time. Summer temperature conditions of 95 F dry-bulb and 60 F wet-bulb are apparently not uncommon in that locality with the Weather Bureau frequently reporting 5 to 10 per cent relative humidities with 95 to 100 F dry-bulb late in the afternoon during days in July and August.

With barometric pressures in Denver of 24.65 in. of mercury it is highly important to consider the differences which exist in vapor pressure, atmospheric moisture content, specific volume and heat content of the air. From charts prepared for this altitude, it is interesting to note that at an air temperature of 84 F saturation one pound of air holds approximately 40 grains more moisture than at sea level.

Air conditioning loads calculated for Denver should also consider the extraordinary amount of solar radiation which is present at high altitudes as the added amounts in heat gain from this source are of prime importance.

Solar Radiation Factors

After the selection of the proper outdoor and indoor temperature conditions, one of the next steps in an analysis of a cooling load is to determine the transmission of heat through the walls, roof and glass. Transmission coefficients are available for almost all of the common building materials and insulators and consequently little variation in final results is obtained from this phase of a cooling load determination.

The solar radiation through walls, roof and glass as used in the various computations of this survey is given in Table 6. It is apparent that many engineers are using the method outlined in THE GUIDE which provides for the addition of a certain number of degrees to the outside dry-bulb temperature for different surfaces. Table 2, appearing on page 160 of THE GUIDE 1937 and based on a number of A.S.H.V.E. papers,[3] was prepared several years ago to provide a simplified method of allowing for sun effect. It is recognized that the calculations for the transmission of heat from solar radiation through building walls are quite complicated due to the periodical character of heat flow and the time lag or heat capacity of construction. Consequently, the present GUIDE method was introduced to be of practical value to the engineer who was unfamiliar with the many variations in solar heat transmission.

Although values of temperature increase for glass are shown in column 23, it should be noted that this method is not suggested in THE GUIDE. Appearing in column 14 are values of solar intensity obtained from curves similar to those on page 161 of THE GUIDE 1937, which should be used with the wall and roof values of columns 22 and 24 for a computation of this form. Percentage values are indicated in column 15 depending upon whether the designer assumed a bare or shaded window.

Occasionally engineers use in addition to the solar intensity and shading factor previously mentioned for glass,

Table 5—Outside and Inside Temperatures for Manila, Philippine Islands

Outside Temp. Deg F		Inside Temp. Deg F	
Dry-Bulb	Wet-Bulb	Dry-Bulb	Wet-Bulb
96	76.0 to 79.0	82	66.0 to 67.0
93	76.0 to 79.0	81	66.0 to 67.0
90	76.0 to 79.0	80	66.0 to 67.0
87	75.5 to 78.5	79	65.5 to 66.5
84	75.0 to 78.0	78	65.5 to 66.5
81	74.0 to 77.0	77	65.5 to 66.5
78	73.0 to 76.0	76	65.0 to 66.0

the heat transmission coefficient and the temperature difference, all as applied to the surface area. This gives an additional heat gain which is unwarranted as the shading factor is a test value which takes into consideration the glass transmission.

In an effort to reduce the selling cost of air conditioning equipment, many manufacturers as well as engineers designing systems have devised empirical methods of computation which incorporate in a single value, factors for heat transmission, and solar radiation. Included in columns 19, 20 and 21 of Table 6 are values of this kind and although admittedly simpler to use, it is readily recognized that they are not based on a scientific approach to the problem. In some cases the values given in these columns are based on the building construction of the job and vary depending upon the conditions stated. When the values are obtained in this manner the results are satisfactory. Considerable engi-

[3] Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932). Absorption of Solar Radiation in Relation to the Temperature, Color, Angle, and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930). Coefficients of Heat Transfer as Measured under Natural Weather Conditions, by F. C. Houghten and C. G. F. Zobel (A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928).

neering judgment is required in using figures of this kind and obviously they should only be used in large applications by an engineer with considerable experience.

In 1935 a paper[4] was presented before the Society which outlined a rational method of cooling load determination based on the use of solar radiation factors with various absorption coefficients when applied to walls facing several directions at different latitudes and at different times of the day. It was evident that six of the calculations submitted were based on this method of computation but with variations in some instances. The values given in columns 11 to 18 inclusive of Table 6, together with the footnotes, show in detail the various items considered.

In the cooling load calculation No. 11 for Urbana, Ill., a composite formula[5] for walls exposed to sun was used which incorporated the sun radiation effects in a single equation together with other sensible heat transmission gain factors as follows:

$$H = \frac{AU}{f_o}\left[\, cI + f_o\,(t_o - t_1)\,\right]$$

where

H = heat transmitted, Btu per hour.
A = wall area, square feet.
U = over-all coefficient of heat transmission, Btu per square foot per hour per degree Fahrenheit temperature difference from air to air.
f_o = outside surface conductance, Btu per square foot per hour per degree Fahrenheit temperature difference between the surface and air.
c = coefficient of absorption of solar radiation.
I = solar intensity normal to surface, Btu per square foot per hour.
t_o = outdoor air temperature, degrees Fahrenheit.
t_1 = indoor air temperature, degrees Fahrenheit.

This equation also takes into consideration the variation in outside surface conductances which the other methods previously described neglected.

Data given in Table 6 show the complete lack of uniformity in the selection of wall surfaces for considering the solar radiation effects. When empirical values were applied it was quite common to select that wall and glass surface having the greatest area irrespective of the compass direction. Other procedures used different solar intensities for the time of day involved as applied to all sun exposed surfaces.

Time Lag and Heat Storage of Walls

Differences in solar intensity values selected for use in the computations were due in some cases to the assumptions made for determining the time of day when the maximum design load occurred.

One customary method of calculation was based on the premise that the changes in load occur coincident with the varying outdoor temperature and sun effect for a particular time of day. Since the walls of a structure offer resistance to heat flow it is obvious that the prevailing outdoor temperature and solar heat are not transmitted instantaneously to the inner surface which introduces large errors when this method is applied to walls having high heat storage capacity.

A few computations took into account time lag assuming sun effects and temperature differences existing at some previous time to determine the maximum load. In one instance, the sun load for the roof was completely discounted as the calculator assumed that this peak load did not occur simultaneously with the peak load on the west wall.

Uncontrolled Infiltration

In general the values stated in Table 6 for determining the amount of uncontrolled air entering the enclosure by infiltration were based on the data given in Chapter 6 of THE GUIDE 1937. In some instances, the calculators used certain established air quantities with surface areas rather than on a basis of the length of crack.

The air change method seemed to be applied to many of the computations varying in amount from half to two per hour and higher according to the judgment of the engineer.

Variations in the amount of infiltration air for the swinging and revolving doors was surprisingly large as is illustrated in column 26 of Table 6. Several values were based on the number of cubic feet of air per passage as directly related to the customer traffic. Others were based on an air quantity per surface area and in a few cases the recommendations were related to the length of door crack. It is apparent that engineers are not in full agreement regarding door infiltration values and with few exceptions little scientific data are available on this subject. A few references in the survey were made to a paper[6] which reported the results of numerous observations conducted on actual installations in the field.

Controlled Ventilation Air

The amounts of outside air introduced for ventilation purposes shown in column 27 of Table 6 ranged principally from 10 to 15 cfm per person with one recommendation approaching a high value of 30 cfm per person. The tabulation indicates that in a few cases the air change method was adopted for establishing the ventilation air requirements.

It was frequently mentioned in the computations that the uncontrolled infiltration requirements were determined on a basis of THE GUIDE method for windows and doors along with the air change method, and then compared with the calculated amount of ventilation air. In almost every case the greatest of the three values was used for determining the total cooling load. Column 28 indicates the various air quantities used in the computation and in a few instances it will be observed that all three items were considered.

Safety Factors

After the total sensible and latent heat quantities were determined several engineers indicated the addition of certain safety factors to take care of the presence of fan motors within the enclosure, duct gains and related items. Heat gain determinations Nos. 11, 15, 16, 33 and 47 allowed directly for fan motors in the amounts of 1 to 3 hp.

[4] Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads by F. H. Faust, L. Levine and F. O. Urban (A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, August, 1935).

[5] Investigation of Summer Cooling in the Warm-Air Heating Research Residence, by A. P. Kratz, M. K. Fahnestock and S. Konzo. (University of Illinois Engineering Experiment Station Bulletin No. 189).

[6] The Infiltration Problem of Multiple Entrances by A. M. Simpson and K. B. Atkinson (A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June 1936).

Table 6—Composite Tabulation of Bank Building Cooling Load Determinations

No.	State	City	Cooling Load Tons of Refrigeration			Safety Factors in Per Cent			Methods of Computing Solar Heat Gain													Heat Gain Allowances for Infiltration and Ventilation Air Entering Enclosure			
									Solar Intensities and Related Factors Used with Surface Areas[a]								Empirical Values Used with Surface Areas			Degrees Added to Outside Dry-Bulb Temp.			Uncontrolled Infiltration		Controlled Outside Air for Ventilation	Column Nos. Used in Computation
			Tot'l	Sensible	Latent	Tot'l	Sensible	Latent	Wall Direction	Wall I	Wall e	Wall F	Glass I	Glass f	Roof I	Roof e	Roof F	Wall	Glass	Roof	Wall	Glass	Roof	Window Air Quantity or Room Air Changes per Hour	Swinging (S) and Revolving (R) Door Air Quantity		
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28
1	Ala.	Birmingham	9.10	8.07	1.03				West									12.8	190	16				40-60 cfh/sq ft	S-60-80 cfh/sq ft R-40-60 cfh/sq ft	20 cfm/person	27
2	Calif.	Los Angeles	12.70	8.95	3.75				West									5.5	100	5.2				1.5 air change		1/3 total air	25+27
3		Los Angeles	7.66	6.95	0.71				West									3.7	180	7.8				2 air change			25
4		Fresno	8.17	8.17	0.00		4		West									3.7	180	7.8				2 air change			25
5		Oakland	7.12	6.44	0.68		4		West									3.7	180	7.8				2 air change			25
6	D. C.	Washington	11.50	8.43	3.07				South West	45 185	0.7 0.7	085 085	20 165		200	0.9	060								320 cfm, based on 200 passages/hour	15% total	26+27
7	Fla.	Miami Beach	6.66	4.66	2.00	10	10	10	South West												13 28	5 5	38		800 cfm total	20 cfm/person	27
8	Ill.	Chicago	7.70	5.71	1.99		10		South West												5 −5	0 43.4	35	20 cfm/sq ft	20 cfm/sq ft	25% total	25+26+27
9		Chicago	7.77	6.73	1.04		10	5	West	50	0.7	09	130		100	0.9	060							15 cfh/sq ft	S-30 cfh/sq ft R-30 cfm x3per x60 = cfh	15 cfm/person	25+26+27
10		Chicago	10.75	9.33	1.42				South West									6.0 0	139 183	60				50 cfh/ft crack using wall having most crack	200 cfh/ft crack for 1 set of doors	15 cfm/person 10% total	27
11		Urbana	9.35	8.25	1.10				South West	90 135	0.7 0.7		90 135	97 97	255	0.9								45 cfh/ft crack using wall having most crack	S-16 cfm/person R-6.5 cfm/person	15 cfm/person	25+26+27
12	Ind.	Indianapolis	9.44	8.04	1.40				South West	45 185	0.7 0.7	087 087	20 165		205	0.9	062							0.78 cfm/ft crack	S-1.56 cfm/ft crack S-8 cfm/person R-6.5 cfm/person	10 cfm/person	25+26
13	Iowa	Des Moines	8.89	7.81	1.08				West										136	27.9				1 air change			25
14	Ky.	Lexington	10.22	7.38	2.84				West				150								20		20	6 air change			25
15	La.	New Orleans	6.47	4.89	1.58		4		West	185	0.7	061	165										5		S-8 cfm/person R-6.5 cfm/person	20 cfm/person	27
16		New Orleans	8.33	6.61	1.72		10		West				80										45	1.5 air change		10 cfm/person	25+27
17	Md.	Baltimore	9.06	7.74	1.32		10	10	South West				100 100								−5 −5		35	0.5 air change		1 air change	25+27
18		Baltimore	8.58	7.13	1.45				South West				60 155								10 10		45	0.5 air change		1 air change	25+27
19	Mass.	Boston	8.01	7.18	0.83		15	10	South West				41 117								0 5		43		S-8 cfm/person R-6.5 cfm/person	10 cfm/person	27
20	Mich.	Detroit	5.91	5.21	0.70		5	5	West									8.2	139					0.78 cfm/ft crack 0.62 cfm/sq ft	1 air change	10 cfm/person	26
21		Grand Rapids	7.75	6.50	1.25		4		West				150								20		20	Infiltration by Guide Method	Not less 1 air change	10 cfm/person	25+27
22		Kalamazoo	8.30	7.34	0.96	10	10		West									5.5	60	9.1						20% total	27
23	Mo.	Kansas City	8.49	7.82	0.60				South West				90 140	97 97							10 20		45			15 cfm/person	27
24		Kansas City	9.24	8.30	0.94				West									3.0	100	12.0						15 cfm/person	27
25	Mont.	Billings	7.57	6.94	0.63				South West									0 0	35 165	13.05						20 cfm/person	27

Table 6—Composite Tabulation of Bank Building Cooling Load Determinations—(Continued)

No.	State	City	Cooling Load Tons of Refrigeration			Safety Factors in Per Cent			Methods of Computing Solar Heat Gain											Heat Gain Allowances for Infiltration and Ventilation Air Entering Enclosure			
									Solar Intensities and Related Factors Used with Surface Areas[a]								Empirical Values Used with Surface Areas			Degrees Added to Outside Dry-Bulb Temp.			Uncontrolled Infiltration		Controlled Outside Air for Ventilation	Column Nos. Used in Computation
			Tot'l	Sensible	Latent	Tot'l	Sensible	Latent	Wall Direction	Wall I	Wall e	Wall F	Glass I	Glass f	Roof I	Roof e	Roof F	Wall	Glass	Roof	Wall	Glass	Roof	Window Air Quantity or Room Air Changes per Hour	Swinging (S) and Revolving (R) Door Air Quantity		
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28
26	Nebr.	Omaha	9.48	7.54	1.94				South West	65 180	0.7 0.7	.087 .087	122 155	68 68	195	0.9	.060									30 cfm/person	27
27	N. J.	Newark	8.46	5.82	2.64	10			South										40	10						25% total	27
28	N. Y.	Buffalo	7.39	5.96	1.43				South West				90 140	68 68							10 20			45	75% of 1.5 air change		15 cfm/person	25+27
29		New York	7.15	5.37	1.78				West												25	25	40			25% total	27
30	N. D.	Grand Forks	8.33	7.27	1.06				South West				80 210								10 20		45	1 air change		1.5 air change	25+27
31	Ohio	Cincinnati	7.04	5.13	1.91				South West				 120								10		45	0.5 air change		1.5 air change	25+27
32		Cincinnati	10.72	6.64	4.08		5		West				79	58							5		45	45 cfh/ft crack	S-17,000 cfh R-3,600 cfh	1 air change	25+26+27
33		Cleveland	8.00	6.63	1.37	10	5		South West				75 100								 15		 25	1 air change		10 cfm/person	25+27
34		Columbus	6.95	5.90	1.05				West				100					.407	1.234	.286	5		30	17 cfh/sq ft or 1.5 air change	90 cfh/sq ft	500 cfm	27
35		Columbus	8.55	7.50	1.05				South West				62 185	97 97							10 20		60	40 cfh/ft crack	R-200 entrances/hour and 40 cu ft/entrance	10 cfm/person	27
36		Dayton	5.90	5.06	0.84				South West									9.0 12.0	45 67	 9.0						10 cfm/person	27
37	Okla.	Oklahoma City	9.00	7.53	1.47				South West				55 207	97 97							11 21		25	2 air change			25
38	Oreg.	Portland	7.19	6.78	0.41		10	5	South West	90 140	0.7 0.7	.085 .085	90 140	97 97	250	0.9	.060									15 cfm/person	27
39	Pa.	Pittsburgh	8.80	7.58	1.22	10			West									3.0	130	12.0				45 cfh/ft crack	S-150 cfh/ft crack R-200 cfh/ft crack	400 cfm	25+26+27
40		Pittsburgh	10.45	8.36	2.09				South West				100 115								10 20		41			25% total	27
41	Tenn.	Memphis	5.23	4.43	0.85				South West									3.0 3.0	30 30	 3.0						10 cfm/person	27
42	Texas	Ft. Worth	8.04	6.33	1.71				West				80								20		45	1.5 air change			25
43	Vt.	Burlington	6.90	6.21	0.69		5	5	West				180								10		30	1.25 air change			25
44	Wash.	Seattle	8.03	7.03	1.00	10	10		South West				80 138	60 60							6 1		31		S-8,400 cfh R-50 pers/hour/door vol.	10 cfm/person not less 20% total	26+27
45	Wis.	Milwaukee	7.99	6.98	1.01	10	10		South West				17 125								0 0		55	45 cfh/ft crack	R-total 165 cfm	10 cfm/person	25+26+27
46	Canada	Winnipeg	7.42	7.04	0.38				South West				73 84								10 0		57			10 cfm/person	27
47		Toronto	10.36	9.79	0.57				South West				161 183								10 0		57	1 air change	S-135 cu ft per pass. for 200 pass/hour R-35 cu ft per pass. for 200 pass./hour	15 cfm/person	26+27

[a] Symbol reference notations: I = solar intensity normal to surface, Btu per square foot per hour; e = coefficient of absorption of solar radiation; F = percentage solar radiation transmitted through wall; f = percentage solar radiation transmitted through bare and shaded windows.

Although not stated specifically, it is probable that some of the safety factors may have been added to take care of starting up loads for the cooling equipment. Standards[7] of one municipality recommended the addition of percentages to the sensible heat load according to the following schedule:

Design Sensible Heat Cooling Loads Tons of Refrigeration	Percentage To be Added	Design Sensible Heat Cooling Loads Tons of Refrigeration	Percentage To be Added
0-1	25	10-14	12
2-4	22	14-16	10
4-6	18	16-20	8
6-8	16	20-25	6
8-10	15	25 and above	5

Most of the factors seemed to be based on personal judgment and there was no value that seemed to be used uniformly. However, there was some evidence obtained from a study of the survey that typical values would be 10 per cent to be added to the sensible heat load and 5 per cent to the latent load. In a few cases an over-all percentage factor was added to the total load.

Summary

With the calculated cooling loads varying from 5.23 to 12.7 tons of refrigeration for the banking problem and with a resulting average of 8.33 tons, it was found impossible to classify the loads for the various cities according to any territorial grouping.

The design outside and inside temperatures used checked closely with recommended values given in THE GUIDE except for the cities mentioned.

A majority of the computations employed THE GUIDE method with variations for determining the solar radiation effects, but several rational procedures were also used which considered actual solar intensities and the nature of the surface.

The greatest variables were found in the methods adopted for estimating the amounts of uncontrolled infiltration and controlled ventilation air. The analysis indicated that a majority of load determinations considered a positive pressure in the enclosure with air introduced in the amounts of 10 to 15 cfm per person.

The results of this survey show clearly that with the different calculation methods now in use it is practically impossible for engineers even in the same locality to appraise the total cooling load on a comparable basis within close limits.

Research Investigations

Observations made in connection with this survey suggest the following research investigations to eliminate several of the variables now existent in a cooling load determination.

1. Study of methods for determining proper outside design air conditions from Weather Bureau records.
2. Continuation of basic physiological studies to determine the correct inside design air conditions for health and human comfort in various types of applications and for specified periods of occupancy.
3. Additional heat transmission tests for new building insulators, glass blocks, and metal building construction.
4. More complete data on heat absorbing glass and shading devices.
5. Additional data regarding effect of heat storage capacity of walls on the time displacement of wall conduction with respect to temperature difference and intensity of solar radiation.
6. More information with reference to effect of heat storage capacity on the magnitude of actual transient heat flow with respect to the magnitude of assumed steady-state flow.
7. Solar radiation absorption characteristics of outside building surfaces in terms of the angle of surface with respect to the rays of the sun.
8. Infiltration tests on new types of metal and wood windows.
9. Investigations of infiltration requirements for swinging and revolving doors.
10. Continuation of the studies for determining the proper ventilation requirements.
11. Initiate a program for determining heat gain from various electrical, gas and steam appliances.
12. Heating effects of artificial illumination and electrical machinery.
13. Studies to check actual cooling requirements of a building with calculated cooling load.

[7] Buffalo Adopts Comfort Air Conditioning Standards (A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September 1936).

Acknowledgment

The preparation of this paper was made possible by the generous assistance of those 50 Society members who were interested in the project and willing to spend the necessary time to prepare a sample calculation. It is hoped that the needed scientific data referred to in this survey will further the investigations of more research in this field and benefit the entire profession as well as those Society members who have contributed data from their personal experience.

Coal Burning Tests on Underfeed Stokers

A study of the characteristics of different coals on the performance of a stoker is analyzed in a report recently released through the Fuel Research Laboratory at the Battelle Memorial Institute, Columbus, Ohio. Although these tests cover only four types of coals, it is indicated that the results are broadly representative of those coals having ash softening temperatures between 2200 and 2400 F.

The effect of size of coal on performance of the small stoker as measured by the possible thermal efficiency of a stoker-fired boiler or furnace, by the tendency to produce smoke, by the extent of coke-free formation, or by the uniformity of the rate of heat liberation was found to be less than had been expected. The measure of performance which was considered of greatest importance, that is the uniformity and dependability of operation, was materially affected by size only with a caking coal.

The greatest effect of the size of coal on the operation of the stokers was the change in the rate of feed due to the difference in the bulk densities of the different sizes. Although one size of coal may perform as well as another when the stoker is properly adjusted, it is important that after the adjustments are made the size be kept as uniform as possible.

The report is titled The Relation of the Size of Bituminous Coals to their Performance on Small Underfeed Stokers and is available through the Bituminous Coal Research, Inc., 803 Southern Building, Washington, D. C., for 50 cents per copy.

Note on the Testing of Aluminum Foil Insulation

By J. D. Babbitt* (*NON-MEMBER*), Ottawa, Canada

RECENTLY during some hot-plate tests on the insulating value of aluminum foil in the *National Research Council*, Ottawa, Canada, it was found that the height of the air space had a very considerable effect on the value obtained for the conductivity. These tests were conducted to determine the insulating effect of aluminum foil when it was used as a curtain to divide an air space into two vertical compartments. In order to measure the conductance of such an arrangement on the hot-plate, test panels were constructed of two wooden frames (outside dimensions 18 x 18 in.) which were built out of strips of wood having a cross section 1 x ½ in. These frames were placed one on each side of the aluminum foil and then firmly joined together with screws. The arrangement is shown in Fig. 1, where the frames have been left open. Before the measurements were made the air spaces were closed by a sheet of brown paper which was firmly glued to the surface of the frames. In this way two closed air spaces were formed each 1 in. wide and separated from one another by the aluminum foil. It was necessary to have one of these panels for each side of the hot plate. Two tests were made: one in which these vertical air cells were left the full area of the frame, and a second in which they were divided into two sections by means of a paper baffle coinciding with the division between the hot-plate and the guard-ring. This baffle is shown in Fig. 1. Two quite different results were obtained from these tests, the former giving a conductance of 0.155 Btu per hour per square foot per degree Fahrenheit, and the latter 0.221. The baffles prevented any exchange of air between the two sections of the air space and were originally intended to prevent any small edge loss due to air movement. The large increase in the conductance could hardly be accounted for by a lateral flow of heat and it was immediately evident that the effect was due to convection currents in the air space.

This problem of the convection currents in vertical air spaces has been treated in several papers[1] and the results obtained indicate that the heat transfer by convection is greater per unit area the smaller the height of the space. Thus Queer specifically states that "blocking thin air spaces decreases the insulating value of the space." Since aluminum foil has become increasingly important as an insulator, it is imperative to emphasize this fact so that tests made at different institutions may be comparable and so that aluminum foil may be used in the most efficient manner. This matter is complicated by the fact that the majority of investigators who have worked with air spaces of the size found in buildings report that the height of an air space has a negligible effect on the heat transfer.[2] Rowley and Algren in their excellent investigation into the conductances of air spaces found no difference in the conductance of an air space when the area was 9 in. square and when it was 3 ft square. The difference in the conclusions arrived at by Queer and by Rowley and Algren appears to be due to the fact that the latter were concerned only with the practical problem of air spaces in actual walls which had emissivity coefficients of about 0.9. In a subsequent paper[3] in which the conductance of air spaces faced with aluminum foil was measured, Rowley did find that the conductance of air spaces 3 ft square was slightly lower than that of air spaces 9 in. square. He stated that "the area of test section may have had some bearing on this difference."

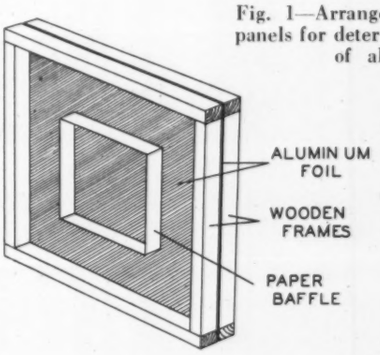

Fig. 1—Arrangement of hot-plate test panels for determining insulating effects of aluminum foil

Since in aluminum foil insulation the greater part of the heat transfer is by convection, this problem of the height of the air space is present in an accentuated form. If the height influences the heat transfer to the extent which Queer found, it is important that some standard height be used for all tests. In order to determine the influence of height of the air space and the nature of convection currents, it was decided to measure the conductances of a series of air spaces of increasing widths and of two different heights.

Description of the Measurements

The measurements were all made on the hot-plate apparatus of the *National Research Council* which has been previously described.[4] It consists of a hot-plate, one foot square, surrounded at all four edges by a guard-ring 3 in. wide. Two identical samples, each 18 in. square, are necessary for a test and these are placed against the surfaces of the hot-plate, one on each side, and are pressed firmly against it by the two cold-plates. Since,

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[1]Ezer Griffiths and A. H. Davis, Dept. Sci. Ind. Res., Food Invest. Board, Special Report No. 9, 1922.
Importance of Radiation in Heat Transfer Through Air Spaces, E. R. Queer, A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932.
R. B. Mason, *Ind. Eng. Chem.*, 25, 245, 1933.

[2]Thermal Resistance of Air Spaces, F. B. Rowley and A. B. Algren, A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929.
M. Fishenden and O. A. Saunders, The Calculation of Heat Transmission, p. 114, 1932.

[3]Insulating value of Bright Metallic Surfaces, F. B. Rowley, A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934.

[4]C. D. Niven, *Can. J. Research*, 7, 115, 1932.

during a test, the hot-plate and guard-ring are maintained at the same temperature, all the heat supplied to the hot-plate must be conducted through the samples and a knowledge of the amount of heat supplied, together with the temperature difference across the samples, is sufficient to determine the conductance. To test the effect of the height of the air space on the conductance it seemed simpler to work with a single air space rather than to continue the measurements originally made on the double air-space arrangement. It was decided, therefore, to confine the measurements to single air cells bounded on both sides with aluminum foil. These air cells were formed by wooden frames similar to those used in the original experiments. The aluminum foil was glued to the two surfaces of the frames, a closed air space being formed in this way. Air spaces of various widths were obtained by changing the thickness of the strips of wood of which the frames were constructed. The width of the strips of wood was throughout ½ in. so that the total area of air space was 289 sq in.

The original measurements had shown quite clearly that the use of a baffle separating the actual test area from the guard-ring made a considerable difference in the heat transfer. In these measurements it was decided to show that the effect of the actual height of the test-area itself was also considerable. As the simplest method of proving this, two series of measurements were made in both of which the test area was separated from the guard-ring by a paper baffle, as shown in Fig. 1. In the first measurements this test area was left as a single air cell, that is, it had a height of 12 in. In the second it was divided into two cells, each 6 in. high, by a horizontal baffle which is represented in Fig. 2. The area of the test area was, therefore, the same in all measurements and any difference in the heat transfer could only be due to the difference in the convection currents under these two sets of conditions.

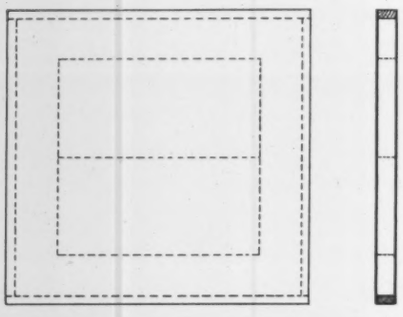

Fig. 2—Diagram of test panel showing horizontal baffle dividing air space

The paper used to construct the baffles had a thickness of 0.007 in. so that the area occupied by the baffles formed a negligible part of the total area and the difference in the conductivity of the paper and the air had no measurable influence on the conductance. That this was actually so was proven by the fact that in a test in which the thickness of the baffles was doubled, no increase in the conductance was noticed. The aluminum foil used was a commercial product which consisted of a layer of kraft paper between two layers of foil, the whole having a thickness of about 0.006 in.

Discussion of Results

The thickness of the sample as measured by the separation of the plates is given in the first column of Table 1.

Table 1—Test Results of Aluminum Foil Insulation with Various Air Cell Heights

Measured Thickness (In.)	Corrected Thickness (In.)	Measured Conductance Btu/Hour/Sq Ft/F Temp. Diff.	Conductance Minus Heat Transmission by Radiation	Heat Transmission by True Conductivity	Heat Transmission by Convection	Temp. Cold Surface, Deg F	Temp. Warm Surface, Deg F	Mean Temp. Deg F
12 in. Air Cells								
0.182	0.170	1.063	1.033	1.010	0.023	28.0	82.6	55.3
0.265	0.253	0.726	0.696	0.690	0.006	28.0	83.4	55.7
0.386	0.374	0.525	0.495	0.467	0.028	28.0	84.0	56.0
0.440	0.428	0.478	0.448	0.406	0.042	28.0	83.8	55.9
0.507	0.495	0.447	0.417	0.350	0.067	28.0	81.3	54.7
0.633	0.621	0.428	0.398	0.279	0.119	28.0	83.3	55.7
0.768	0.756	0.449	0.419	0.230	0.189	28.0	81.3	54.7
1.012	1.000	0.459	0.429	0.174	0.255	28.5	80.1	54.3
1.227	1.215	0.420	0.390	0.143	0.247	28.0	84.3	56.1
1.532	1.520	0.421	0.391	0.113	0.278	28.0	82.5	55.2
1.994	1.982	0.425	0.395	0.088	0.307	28.0	83.2	55.6
6 in. Air Cells								
0.191	0.179	1.051	1.021	0.973	0.048	28.0	84.7	56.3
0.269	0.257	0.721	0.691	0.680	0.011	28.0	84.2	56.1
0.392	0.380	0.539	0.509	0.453	0.056	28.0	83.6	55.8
0.447	0.435	0.513	0.483	0.400	0.083	28.0	81.5	54.8
0.505	0.493	0.501	0.471	0.351	0.120	28.0	81.5	54.8
0.636	0.624	0.483	0.453	0.277	0.176	28.0	81.2	54.6
0.770	0.758	0.507	0.477	0.230	0.247	28.0	82.5	55.2
1.006	0.994	0.508	0.478	0.176	0.302	28.0	82.5	55.2
1.228	1.216	0.486	0.456	0.142	0.314	28.0	80.8	54.4
1.533	1.521	0.468	0.438	0.113	0.325	28.0	81.3	54.6
1.998	1.986	0.457	0.427	0.088	0.339	28.0	83.7	55.7

This is, of course, the width of air space plus aluminum foil and to obtain the width of the air space alone, the thickness of the two sheets of aluminum foil must be deducted. The resultant width is given in the second column. The next column includes the actual measured conductance in Btu per hour per square foot per degree Fahrenheit temperature difference between the two surfaces. This is made up of the heat transferred by the three methods, conduction, radiation, and convection. It is possible to compute the quantity of heat transferred by conduction and radiation as distinct from convection. The true conductivity of air has been measured by hot wire and horizontal plate methods and the transfer by radiation can be calculated by Stefan's law, if the temperatures and emissivities of the two radiating surfaces are known. Since it may be assumed without sensible error that the aluminum foil is at the same temperature as the plates and since the emissivity of the aluminum foil in this range has been determined by several observers[5] and may be taken as approximately equal to 0.05, it is possible to calculate the heat transferred by radiation. When these two quantities are subtracted from the measured conductance, the result is the heat transferred by convection. This value is given in the sixth column of Table 1. The last three columns give the temperatures of the hot and cold surfaces and the mean temperature respectively.

These results have been plotted in Fig. 3, where the heat transfer (Btu per hour per square foot per degree Fahrenheit temperature difference) is represented as a function of the width of air space (in.). The dots and circles give the measured conductance of the 12 in. air

[5]Loc. Cit. p. 19, See Note 2.

cells while the crosses give the same data for the 6 in. cells. It is to be noticed that there is a distinct but small maximum and minimum in the curves. It is also interesting to notice that the two curves practically coincide at widths less than 0.4 in. This is strong evidence that below this width convection plays no part in the heat transfer. This should be expected when considering the mechanism of heat transfer by convection currents in

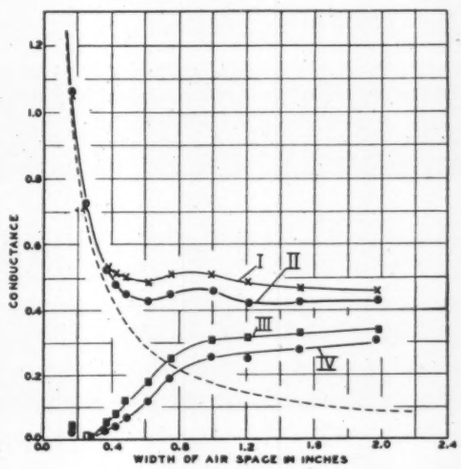

Fig. 3—Relation between conductance and width of air space

Curve I: Conductance of 6 in. air cells
Curve II: Conductance of 12 in. air cells
Curve III: Heat transfer by convection alone in 6 in. cells
Curve IV: Heat transfer by convection alone in 12 in. cells
Dotted curve represents the heat transfer by true conduction.

these vertical air spaces. There must be a current of air circulating in the air space, going down the cold side across the bottom, up the warm side and back across the top. There must be a certain minimum distance between the plates necessary to allow the free movement of these currents; at smaller distances the upward and downward currents would interfere and eventually with decreasing width there can be no currents at all. In these experiments this evidently takes place around 0.4 in.

The value obtained for the conductivity of air when it is measured in such a manner as to eliminate both convection and radiation is 5.77×10^{-5} calories per square centimeter per second per degree Centigrade temperature difference per centimeter.[6] The dotted curve in Fig. 3 gives the conductance of air spaces of varying widths calculated from the air conductivity value as corrected for a mean temperature of 55 F according to the temperature coefficient given in the International Critical Tables.

Curves III and IV in Fig. 3 represent the heat transferred by convection alone, which were obtained by subtracting the heat transferred by conduction and radiation (obtained as explained previously) from the measured conductances. The dots and squares represent the values for the two 6 in. air cells and the crosses and circles those for the 12 in cells. It is seen in both cases that around 1 in. the heat transferred by convection starts to decrease and becomes zero at about 0.3 in. According to other measurements[7] the convection currents near hot or cold vertical surfaces are confined to the air layers within 1 in. of the surface and the fact that heat transfer in these tests does not begin to decrease until the width has been decreased to 1 in. confirms this fact. This is the maximum distance at which the convection currents interfere, which seems to indicate that there is very little air movement at distances greater than ½ in. from the surface.

These measurements clearly substantiate Queer's results and show that the height of the air space must be taken into consideration in all tests of aluminum foil insulation. It is very desirable that some standard height be assigned for these measurements on air spaces and it would seem logical to choose one foot as the standard height since the conductance is generally given in terms of Btu per square foot. If such standard practice is not feasible, because of the variation in the size of hot-plates, it is imperative that the height of the air space should be given in all reports of such tests.

[6] Reports on Progress in Physics, *The Physical Society*, II, 196, 1935.
[7] Loc. Cit. See Note 1.

Engineering Registration Laws

The legal registration of professional engineers is now required in 34 of the United States and in the territories of Hawaii and Puerto Rico. Illinois only requires the registration of structural engineers and Texas makes provisions for the licensing of land surveyors. All states and territories which have registration laws, with the exception of Colorado, are members of the National Council of State Boards of Engineering Examiners, who have continually stressed the desirability of uniformity in the drafting of licensing laws.

As a result of a recent series of conferences at which representatives of various engineering societies were present, a revision of a model law drafted in 1932 was completed. This revision was not a drastic change from the previous model law but included such desirable modifications which were intended to make it once more fully abreast of the times. Copies of the revised draft have been submitted to the governing groups of the various engineering societies for consideration and action, after which the law will be available for general distribution.

Representatives who were responsible for the recent revisions to the model law include individuals from the following engineering societies: *American Society of Civil Engineers, American Institute of Mining and Metallurgical Engineers, American Society of Mechanical Engineers, American Institute of Electrical Engineers, American Association of Engineers, American Engineering Council, American Institute of Chemical Engineers, American Institute of Consulting Engineers, American Society of Heating and Ventilating Engineers, American Society of Safety Engineers, Connecticut Society of Civil Engineers, Engineers' Council for Professional Development, Institute of Radio Engineers, National Council of State Boards of Engineering Examiners, National Society of Professional Engineers,, New York Association of Consulting Engineers* and *Society of Naval Architects and Marine Engineers.*

Non-Dimensional Fan Characteristics

By H. Carlton Moore* (*MEMBER*), Cambridge, Mass.

WHEN perusing fan manufacturers' catalogs, one is impressed with the large number of tabulated values giving speed, horsepower, efficiency, etc., for each different size of each series of geometrically similar fans manufactured by the company. Even when in the form of performance curves, there is usually a number or series of curves which have to be read completely in order to find the required data.

An attempt has been made in this discussion to present in non-dimensional ratios a comparatively simple set of three curves, which will furnish all of the information required for any one series of geometrically similar fans. This set of curves would save many pages of tabulated values. Moreover, it would allow the fan manufacturer to test completely only one size of fan and predict the performance with accuracy of any other fan of similar geometrical dimensions.

The data from which the curves were plotted were not from tests made by the author, but taken from standard fan manufacturers' catalogs at random, and checked over a wide range of sizes and speeds. Of course, each style of fan will have a different set of non-dimensional curves, but after once having been established for that particular style, they will hold for any size or speed, if the dimensions are geometrically similar.

The non-dimensional ratios used, do not consist of any complicated quantities, but quantities familiar to the ordinary ventilating engineer. They are defined as follows:

σ = ratio of static or resistance head to velocity head.

$\phi = \frac{V}{\omega r}$, where V is the air velocity (probably more convenient to use the exit velocity) and ωr is the tip speed of the rotor.

η = fan efficiency.

λ = power coefficient (calculation shown later).

As long as the dimensions of the numerator and denominator are the same, the units are immaterial.

The power output in foot pounds per minute of any fan is usually calculated as the product of the weight of air multiplied by the total head in feet of air, or algebraically:

Power Output = $(Q \times d) \times$ (T.H.)

where Q = cubic feet of air per minute.

T.H. = total head in feet of air (resistance head plus velocity head).

d = air density in pounds per cubic foot.

However, Q equals the product of the exit area by the exit velocity, and the exit velocity equals $\phi(\omega r)$. Also, T.H. equals velocity head multiplied by $(1 + \sigma)$. Now, the velocity head is $\frac{V^2}{2g \times 3{,}600}$ if V is in feet per minute.

Again, $\phi^2(\omega r)^2$ may be substituted for V^2. So by substitution of these values:

Power output = $A\,\phi^3\,(\omega r)^3\,(1 + \sigma)\,\frac{d}{2g \times 3{,}600}$, and

the power required to drive is the power output divided

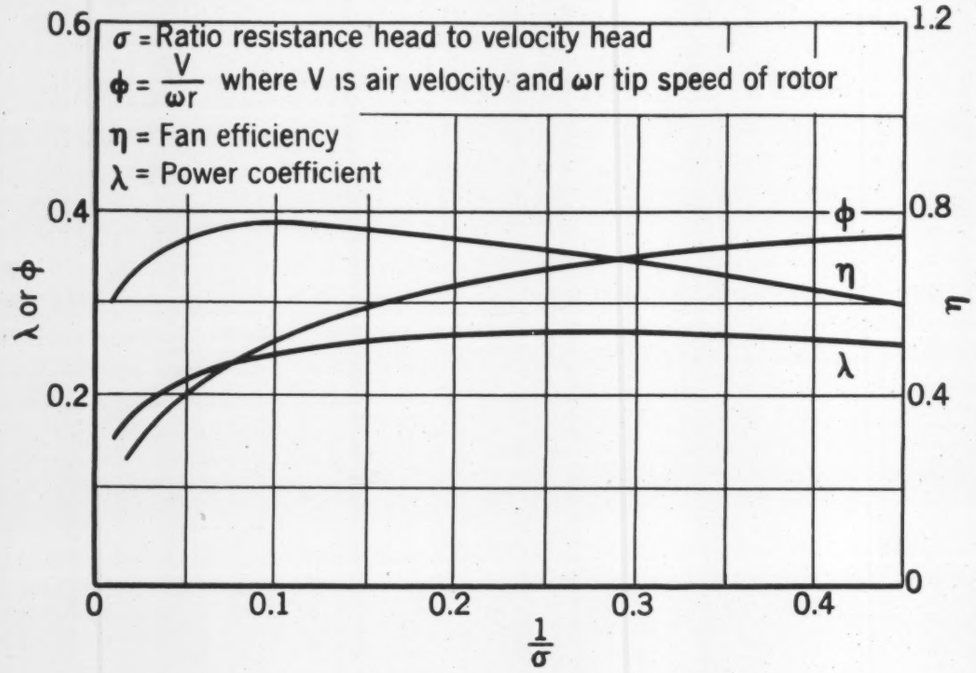

Fig. 1—Non-dimensional performance curves for backward curve blade fan

*Instructor in Mechanical Engineering, Massachusetts Institute of Technology.

by the fan efficiency. So by letting $\lambda = \frac{(1+\sigma)\phi^3}{\eta}$, the horsepower required to drive the fan is simply:

$$\text{Horsepower} = \frac{\lambda A (\omega r)^3 d}{2g \times 3{,}600 \times 33{,}000}$$

where A = outlet area in square feet.

(ωr) = tip speed in feet per minute.

$\frac{d}{2g} = 0.001165$ (for 70 F air).

By the use of the λ curve, the horsepower required to drive may be calculated as $9.82 \times 10^{-12} \lambda A(\omega r)^3$ where the symbols have the same meaning as in the previous equation.

A set of curves is shown in Fig. 1 for a backward curve blade fan, while Fig. 2 shows a set for a forward curve blade type fan, both plotted in non-dimensional ratios as shown.

As an example, assume that a fan of the style represented by Fig. 1 was tested, resulting in the curves shown, and also that the tested fan had the following dimensions: diameter of rotor, 12 in.; diameter of suction duct connection, 12 in.; and, discharge opening, 10 x 12 in.

Assume that it is desired to select a fan of this series to furnish 36,000 cfm at 70 F, at a static pressure of 2.00 in. of water and a velocity pressure of 0.20 in. of water.

By calculation $\sigma = 10$ or $\frac{1}{\sigma} = 0.10$. Referring to the curves, where $\frac{1}{\sigma} = 0.10$, $\phi = 0.262$, which is the ratio of the air discharge velocity to the tip speed. The velocity corresponding to a velocity pressure of 0.20 in. of water is, according to the standard formula, $1{,}096\sqrt{\frac{\text{V. P.}}{d}}$ or equals $1{,}096\sqrt{\frac{0.20}{0.075}} = 1{,}794$ fpm in this case. Immediately the discharge area required is 36,000 divided by 1,794 or 20.1 sq ft. The nearest size which would be commercially made having geometrically similar dimensions would probably be 50 x 60 in. or approximately 20.8 sq ft. These would be five times the dimensions of the original test fan, so the diameter of the rotor would be 5 ft. The tip speed would be the velocity divided by ϕ, or 1,794 divided by 0.262 which equals 6,840 fpm. This corresponds to πDN fpm, or

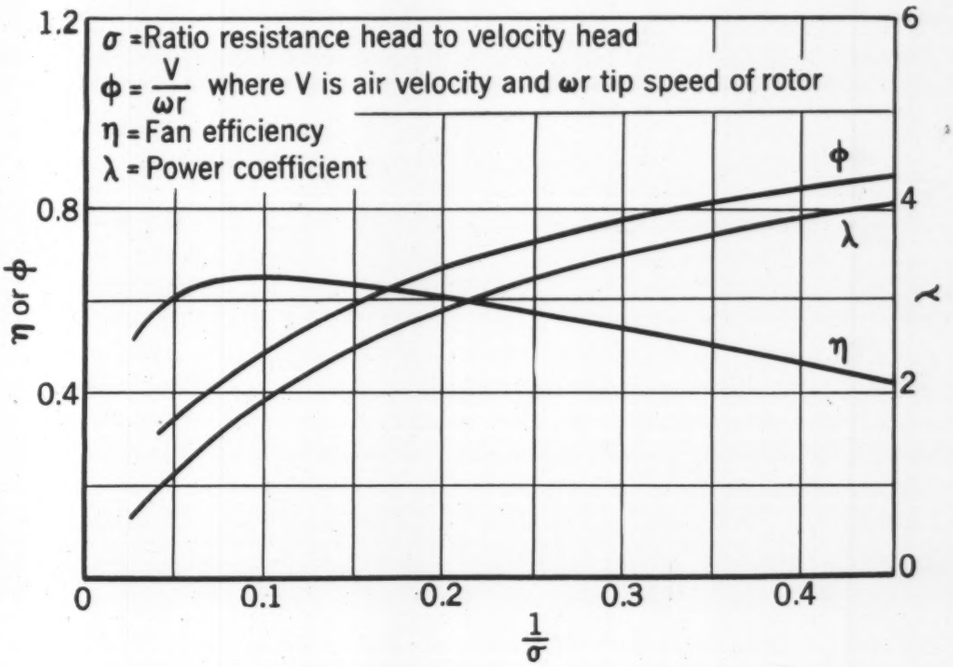

Fig. 2—Non-dimensional performance curves for forward curve blade fan

$N = \frac{6{,}840}{\pi \times 5} = 435$ rpm. The horsepower required to drive this fan would be $9.82 \times 10^{-12} \lambda A(\omega r)^3$, where $\lambda = 0.254$ from the curve where $\frac{1}{\sigma} = 0.10$, or

Horsepower $= 9.82 \times 10^{-12} \times 0.254 \times 20.8 \times (6{,}840)^3 = 16.6$.

It is also readily noted from the curves that this fan is operating very near its maximum efficiency.

Transactions Ready for Distribution

A.S.H.V.E. TRANSACTIONS, Volumes 41 and 42, will come from the press September 15 and distribution will commence. All members of the Society whose dues have been fully paid for the years 1935 and 1936 will receive copies. Work of compiling the 1937 volume is now under way.

Reprints of A.S.H.V.E. Papers

There are available in the files of the Society, many reprints of papers which have appeared in its JOURNAL and TRANSACTIONS. For those who wish copies they can be supplied at a nominal cost of 10 cents each for Series 4-100 to 28-820 (1904-1928) and 15 cents each for Series 28-818 to 36-060 (1928-1936).

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W. A. Danielson, *Chairman*.
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• Abnormal Air Conditions in Industry; Effects on Workers and Methods of Control, by C. P. Yaglou. *Journal of Industrial Hygiene and Toxicology*, vol. 19, January, 1937, pp. 12-43.

• Prevention of Furnace Explosions, by E. G. Bailey. *Power Plant Engineering*, vol, XLI, no. 8, August 1937, pp. 464-466.

• The Air Resistance of Passenger Trains, by F. C. Johansen. *Proceedings of the Institution of Mechanical Engineers* (London), vol. 34, 1936, pp. 91-208. Comprehensive study of air resistance, including effects of oblique winds, carried out in a wind tunnel on reduced-scale models of passenger rolling stock. Aerodynamic forces measured separately. Study of streamlining shows 75 per cent of air resistance of fast trains can be saved by streamlining. Paper gives analysis of influence of direction of natural winds.

• The Resistance of the Flow of Water Along Smooth Rectangular Passages, and the Effect of a Slight Convergence or Divergence of the Boundaries, by J. Allen and N. D. Grundberg. *Philosophical Magazine* (London), vol. 23, no. 154, March 1937, pp. 490-503. Description of test equipment to study flow of water in rectangular passages. Data obtained and mathematical interpretation of results.

• The Physicist-Engineer in Research and Industry, by Dayton Ulrey. *The Journal of Engineering Education*, vol. XXVII, no. 10, June 1937, pp. 782-783. Discussion of college training of physicists to prepare them for industrial research engineers. Characteristics of the successful physicist-engineer which the university can do much to develop, especially during the graduate period.

• Power Production from Tropical Vegetable Waste, by Xaver Soehner. *Engineering* (London), vol. CXLIII, no. 3723, May 21, 1937, pp. 569-571, and May 28, pp. 599-600. Description of suction gas plant for the gasification of tropical vegetable waste. The furnace is designed to burn rice chaff, cotton seed coats, coconut shells, or any other vegetable waste. Diagram of three systems of gasification; (1) rising flow gas with production of tar, (2) a descending gas flow with the partial production of unburnt tar, (3) the double flow system giving tarless gasification.

• Ventilating a Large Underground Garage, by Joseph F. Kern. *Heating and Ventilating*, vol. 34, no. 7, July 1937, pp. 35-38. Description of ventilation system in a 200-car garage under traffic circle at 79th St. on Hendrick Hudson Drive, New York City. Dual exhaust system used. Four entrances to garage are air inlets; air outlets in ceiling.

• Thermal Conductivity of Materials Used for Solid Carbon Dioxide Containers, by Ezer Griffiths and J. H. Awbery. *Ice and Refrigeration*, vol. 92, no. 6, June 1937, pp. 393-395. De scription of test and data on thermal conductivity of cork, expanded rubber and balsa wood to be used as insulation in solid carbon dioxide containers. Thermal Conductivity found to vary as cube root of density of above insulating materials.

• Some Factors Affecting the Design of Heat Transfer Apparatus, by E. W. Still. *Proceedings of the Institution of Mechanical Engineers* (London), vol. 134, 1936, pp. 363-435. Paper indicates solution of heat transfer problems in simplest manner. Author endeavors to show that much of the information on film heat flow, for turbulent conditions of liquids and gases in and across pipes, can be expressed by one equation. Theory by which film rates of heat flow can be obtained from overall rates of heat transfer; based on this theory, tests with water, ethylene glycol, oils and air have been analyzed. New data for streamline flow. Air flow across tube nests and over grilled surfaces is considered. Object of paper is to aid design of heat transfer apparatus.

• Thermal Conductivity of Building Materials, by Frank B. Rowley and Axel B. Algren. *Bulletin No. 12, Engineering Experiment Station, University of Minnesota*, 134 pages. Report of tests made at University of Minnesota, and in cooperation with the Research Laboratory of the A.S.H.V.E., on thermal conductivity of building materials. Description and illustration of test equipment; data on insulating material and built-up walls; also masonry walls.

• Car Cooling and Immersion Cooling, by R. W. Randall. *Ice and Refrigeration*, vol. 92, no. 6, June 1937, pp. 429-430. Discussion of value of precooling fruit and vegetable shipments. Precooling preserves fruits longer and allows picking at riper stage, therefor making it possible to deliver better products.

• Household Decorations and Repairs. *English Mechanics* vol. XXII, no. 555, p. 210, June 1937. Description of practical methods of making house repairs. Detailed illustrations of methods of stopping water leaks around and under outside doors. A simple and artistic method of hanging curtains with two or more rods and an outside wooden framework.

• How to Build to Save Fuel, by G. L. Larson. *American Builder and Building Age*, vol. 59, no. 6, June 1937, pp. 76-77. Discussion of value of insulation, plus storm doors and sash in nine types of residence construction in terms of fuel consumed, and original heating plant cost.

• Pipe Corrosion and Coatings—Corrosion Survey, by Erick Larson. *American Gas Journal*, vol. 146, no. 6, June 1937, pp. 63-66. Corrosion survey is a recorded examination of the effects of corrosive forces on the physical condition of a metal. Discussion of conditions of survey, time and frequency of observations, and classification of corrosion by appearance.

• Preventing Corrosion, by H. G. Lloyd. *Illustrated Official Journal of Patents* (London), 1936, Group 18. Pipes are protected from corrosion by a deposit of calcium carbonate on the surface. Method may be applied to pipes containing flowing water.

• Fluid-Flow Analyser, by H. L. Parr. *Mechanical Engineering*, vol. 59, no. 6, June 1937, pp. 437-439. A small inexpensive device that is simple to operate. Description of fluid-flow analyzer designed to illustrate principles of fluid dynamics to engineering students. Model of new apparatus may be placed in analyzer to determine air flow; if flow is not satisfactory, model may be adjusted for improvement.

• Notes on the Measurement of Air Flow, by E. Ower. *Colliery Engineering*, vol. 14, no. 160, June 1937, pp. 287-290. Discussion of methods available for measuring air flow, the conditions in which each can be used, and its practical limitations. Also the problem of measuring pulsating flow of air. Methods listed are pressure tube and constriction anemometers; mechanical type, both vane and plate, and hot-wire anemometers.

• Combating Dust in the Refractory Industry, by Werner Faust. *Tonind-Ztg*, vol. 61, no. 6, pp. 59-61, and no. 7, pp. 74-76, 1937. Suction apparatus and other devices for eliminating dust and safeguarding the health of workers. Masks worn by the workers as illustrated.

• Future Heating from a Research Angle, by Charles W. Good. *Coal-Heat*, vol. 31, no. 6, June 1937, p. 21. Examples of several research projects which have been valuable to sponsors of research at the University of Michigan.

Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS *is located*

COMMITTEE ON RESEARCH

W. A. DANIELSON, *Chairman;* W. L. FLEISHER, *Vice-Chairman;* F. C. HOUGHTEN, *Director;* DR. A. C. WILLARD, *Technical Adviser;* A. C. FIELDNER, *Ex-Officio Member. One year:* C. A. DUNHAM, W. L. FLEISHER, ELLIOTT HARRINGTON, A. P. KRATZ, H. C. MURPHY. *Two Years:* W. A. DANIELSON, C. E. LEWIS, D. W. NELSON, C. TASKER, C.-E. A. WINSLOW, *Three Years:* H. E. ADAMS, A. E. STACEY, G. L. TUVE, J. H. VAN ALSBURG, J. H. WALKER.

Executive Committee: W. A. Danielson, *Chairman;* W. L. Fleisher, J. H. Walker.

Technical Advisory Committees

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Committee on Air Conditioning Requirements of Glass—IF-18: M. L. Carr, *Chairman;* F. L. Bishop, A. N. Finn, E. H. Hobbie, R. J. Lillibridge, R. A. Miller, F. W. Parkinson, W. C. Randall, L. T. Sherwood, J. T. Staples, C. Tasker,* G. B. Watkins, F. C. Weinert.

Committee on Air Distribution—IP-21: Ernest Szekely, *Chairman;* S. H. Downs, M. K. Fahnestock, F. J. Kurth, D. W. Nelson,* C. H. Randolph, J. E. Schoen, G. L. Tuve,* J. H. Van Alsburg.*

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Committee on Climate and Air Conditioning—C-24: Dr. C. A. Mills, *Chairman;* Major G. C. Dunham, James Govan, Dr. W. J. McConnell, C. F. Neergaard, Dr. F. M. Pottenger, Jr., E. L. Weber, Prof. C.-E. A. Winslow.*

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Committee on Corrosion in Air Conditioning Equipment—IF-14: A. E. Stacey, Jr.,* *Chairman;* A. F. L. Anderson, M. L. Diver, F. L. LaQue, C. E. Lewis,* R. M. Palmer, F. N. Speller, C. M. Sterne, R. T. Thornton, J. H. Young.

Committee on Corrosion in Steam Systems—IF-2: J. H. Walker,* *Chairman;* W. H. Driscoll, C. A. Dunham,* L. B. Miller, R. R. Seeber, F. N. Speller, C. M. Sterne.

Committee on Effect of Water on Roofs—IF-29: A. B. Snavely, *Chairman;* M. R. Beasley, J. B. Griffiths, Elliott Harrington,* E. H. Hyde, W. L. Murray, E. R. Queer, C. S. Reeve, E. T. Selig, Jr.

Committee on General Air Conditioning Comfort Requirements—OH-5: C. P. Yaglou, *Chairman;* J. J. Aeberly, R. R. Sayers, C.-E. A. Winslow.*

Committee on Heat Requirements of Buildings—IF-8: O. W Armspach, *Chairman;* P. D. Close, W. H. Driscoll, H. M. Hart, V. W. Hunter, H. H. Mather, E. C. Rack, F. B. Rowley, R. J. J. Tennant, J. H. Walker.*

Committee on Heat Transfer of Finned Tubes with Forced Air Circulation—IP-10: F. B. Rowley, *Chairman;* H. F. Hutzel, R. H. Norris, C. H. Randolph, W. E. Stark, G. L. Tuve,* C. F. Wood.

Committee on Insulation—IF-23: L. A. Harding, *Chairman;* E. A. Allcut, H. C. Bates, H. C. Dickinson, J. D. Edwards, E. C. Lloyd, W. E. McMullen, R. T. Miller, E. R. Queer, T. S. Rogers, F. B. Rowley, W. S. Steele, C. Tasker,* B. Townshend, G. B. Wilkes.

Committee on Intermittent Heating—IP-20: E. K. Campbell, *Chairman;* W. L. Cassell, Prof. E. F. Dawson, N. W. Downes, F. E. Giesecke, J. M. Robertson, J. H. Kitchen, Prof. A. H. Sluss, G. L. Tuve.*

Committee on Psychrometry—C-11: F. R. Bichowsky, *Chairman;* C. A. Bulkeley, J. A. Goff, Dr. E. V. Hill, F. G. Keyes, A. P. Kratz,* W. M. Sawdon.

Committee on Radiation with Gravity Air Circulation—IP-9: M. K. Fahnestock, *Chairman;* B. C. Benson, H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, D. W. Nelson,* R. N. Trane, T. A. Novotney.

Committee on Relation of Body Changes to Air Changes—OH-3: Dr. E. V. Hill, *Chairman;* N. D. Adams, J. J. Aeberly, John Howatt, A. P. Kratz,* P. J. Marschall, V. L. Sherman.

Committee on Sound Control—IF-1: J. S. Parkinson, *Chairman;* C. M. Ashley, G. F. Drake, V. O. Knudsen, R. F. Norris, C. H. Randolph, J. P. Reis, A. E. Stacey,* G. T. Stanton, F. R. Watson.

Committee on Summer Air Conditioning for Residences—IP-7: M. K. Fahnestock, *Chairman;* E. A. Brandt, John Everetts. Jr., Elliott Harrington,* H. F. Hutzel, E. D. Milener, K. W. Miller, E. B. Newill, F. G. Sedgwick, J. H. Walker.*

Committee on Transportation Air Conditioning—C-12: L. B. Miller, *Chairman;* T. R. Crowder, F. B. Rowley, A. E. Stacey, Jr.,* L. W. Wallace.

Committee on Treatment of Air with Electricity—C-17: Prof. C.-E. A. Winslow,* *Chairman;* R. D. Bennett, W. H. Carrier, L. W. Chubb, Major W. D. Fleming, R. F. James, L. R. Koller, Dr. C. A. Mills, Prof. E. B. Phelps, Prof. G. R. Wait, Prof. W. T. Wells.

Committee on Weather Design Conditions—IF-31: W. E. Stark, *Chairman;* E. W. Goodwin, A. C. Grant, J. H. Kincer, A. P. Kratz,* L. S. Ourusoff.

*Member of Committee on Research.

A. S. H. V. E. RESEARCH

A Page of News Notes and Comment

By **Brewster S. Beach***

1938 Nominees to Committee on Research

THE following have been nominated by the Council to membership on the Committee on Research for a 3-year term beginning in 1938: M. K. Fahnestock, W. L. Fleisher, R. J. Tenkonohy, T. H. Urdahl, Baldwin M. Woods.

Maurice K. Fahnestock, M.E., B.S., M.S., Research Asst. Prof., Dept. Mech. Eng., University of Illinois, Urbana, Ill.

Prof. Fahnestock is known throughout the air conditioning, heating and ventilating industry for his brilliant work at the Engineering Experiment Station of the University of Illinois in association with Prof. A. P. Kratz, Research Professor and Seichi Konzo. He is an authority on steam and hot water heating, ventilating and air conditioning and author of many bulletins and articles on these subjects. He has been associated with the University of Illinois, from which he graduated in 1924, for the past eleven years. He is a member of the *A.S.M.E.* as well as of the A.S.H.V.E.

W. L. Fleisher, B.S., M.S., M.E., Consulting and Advisory Engineer, New York, member of the Council, A.S.H.V.E.

Mr. Fleisher has been nominated to succeed himself on the Committee on Research. He is one of the most active members of the Society devoting special attention to the development of the research activity and is Vice-Chairman of the Committee on Research. He is a graduate of the University of Pennsylvania, inventor of many devices and systems used in air conditioning and the author of numerous scientific papers. Mr. Fleisher is a member of the *A.S.M.E.*, *A.S.R.E.* and other organizations. He was one of the prime movers in bringing out the first A.S.H.V.E. Guide and was Chairman of the Guide Publication Committee 1933-34.

R. J. Tenkonohy, B.S., vice-president, Airtherm Manufacturing Co., St. Louis, Mo.

Mr. Tenkonohy has been a member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS since 1923. He graduated from Michigan Agricultural College in 1912, with the degree of B.S. after receiving his preparatory educational training at the Detroit Technical Institute. He is a native of Detroit. Upon his graduation he was connected with the American Blower Corp., Skinner Brothers Mfg. Co. and Shannon Mackintosh Co. of Detroit. He is an able sales engineer-executive and is vice-president of the Airtherm Manufacturing Co., St. Louis, Mo. Mr. Tenkonohy is also a member of the *Detroit Engineering Society*, secretary of the St. Louis Chapter, A.S.H.V.E. in 1936 and now is 2nd vice-president.

Thomas H. Urdahl, Consulting Engineer, president, Engineering Management, Inc., Washington, D. C.

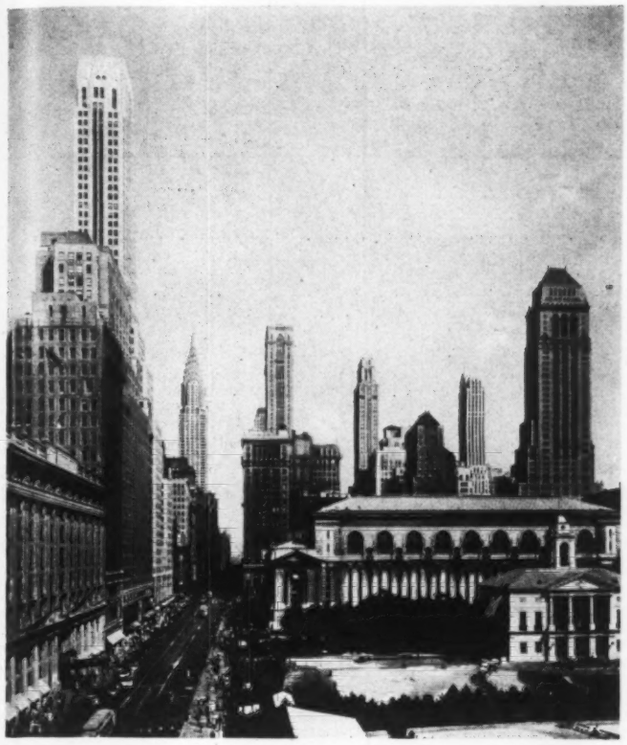
Mr. Urdahl is an active member of the Society, and among the best known consulting engineers in the City of Washington. He is a member of the Board of Governors of the Washington D. C., Chapter, A.S.H.V.E., attended Swarthmore College, majoring in mechanical engineering, and carries on his own professional practice, specializing in combustion, heating, ventilating, air conditioning and the mechanical equipment of buildings, including sanitary and electrical engineering. Mr. Urdahl is a member of the *American Association of Engineers*, past president of its Washington Chapter, the *National Educational Association* and other organizations.

Baldwin M. Woods, E.E., M.S., Ph.D., Chairman, Department Mechanical Engineering, University of California, Berkeley, Calif.

Prof. Woods is a distinguished addition to the Committee on Research. He was born in Texas in 1887 and is a graduate of the University of Texas where he received the degree E.E. He took his master's degree of M.S. and Ph.D. at the University of California, and studied in Paris and Munich. Prof. Woods' chief interests are aeronautics, mechanics and air conditioning. Listed in "Who's Who in America," and "Who's Who in Engineering." Prof. Woods is the author of many scientific papers. He is president of the new Golden Gate Chapter of the A.S.H.V.E., member of *A.S.M.E.*, *Institute of Aeronautical Sciences*, Sigma Xi and Fellow of the *American Association for the Advancement of Science*.

*Director, Information Service, Committee on Research.

44th Annual Meeting of A. S. H. V. E.



Forty-Second St., New York

The Heating and Ventilating Exposition and Air Conditioning Show

EVERYONE engaged in heating, ventilating and air conditioning will be attracted to New York City during the week of January 24, 1938, because of the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the Annual Meeting of the *American Society of Refrigerating Engineers,* the Mid-Winter meeting of the *National Warm Air Heating and Air Conditioning Association* and the 5th International Heating and Ventilating Exposition.

Members of the A.S.H.V.E. will be entertained by the New York Chapter and Pres. W. E. Heibel has announced the appointment of a Committee on Arrangements with A. C. Buensod, Chairman, and Alfred Engle, Vice-Chairman. Headquarters for the meeting will be in the Hotel Biltmore and sessions will be held from Monday to Friday.

The *American Society of Refrigerating Engineers* and the *National Warm Air Heating and Air Conditioning Association* have selected the Hotel Roosevelt as headquarters.

This fourth week of January will provide an opportunity for engineers to meet and exchange ideas, discuss the trend of engineering development and practices and enjoy a fine display of modern equipment.

Plans should be made now to be in New York for the entire week of January 24th. Mark the calendar and reserve the dates.

The Exposition

On September 1 when a total of 230 firms had engaged space in Grand Central Palace, New York for the Fifth International Heating and Ventilating Exposition, January 24 to 28, 1938, it was evident that three floors of the Palace would be required to house the show. All of the space on the first and second floors has been taken and indications are that this fifth show will establish an all-time record from every standpoint.

Previous expositions have been held in Chicago, 1936, New York, 1934, Cleveland 1932 and Philadelphia 1930 under the personal direction of C. F. Roth, who is managing the 1938 show.

Those who desire a complete picture of the engineering developments, in all of the diverse phases of construction, will find the mechanical equipment exhibited at this Exposition of special interest. Every division of air conditioning, cooling, heating, piping and ventilating of homes, offices, factories and public buildings will be represented. Automatic equipment and accessories are to be featured. Coal, oil and gas burning apparatus will be shown. There will be a comprehensive drafting of tanks, pumps, valves, regulators, unit heaters, motors, fans, registers, grilles, and every product of interest to engineers, architects, and operators of factories, public buildings, theaters, and ocean liners.

The Exposition will tell the whole story of modern heating, ventilating and air conditioning. See the 1938 models for the home and office of the future.

NEWS OF LOCAL CHAPTERS

OFFICERS OF LOCAL CHAPTERS—1937

ATLANTA: *Organized,* 1937. *Headquarters,* Atlanta, Ga. *President,* E. W. KLEIN, 152 Nassau St., N. W. *Secretary,* C. T. BAKER, 713 Glenn St., S. W.

CINCINNATI: *Organized,* 1932. *Headquarters,* Cincinnati, O. *Meets,* Second Tuesday in Month. *President,* I. B. HELBURN, 610 Chamber of Commerce Bldg. *Secretary,* H. E. SPROULL, 1005 American Bldg.

GOLDEN GATE: *Organized,* 1937. *Headquarters,* San Francisco, Calif. *President,* B. M. WOODS, Univ. of Calif., Berkeley, Calif. *Secretary,* G. J. CUMMINGS, 113 Tenth St., Oakland, Calif.

ILLINOIS: *Organized* 1906. *Headquarters,* Chicago, Ill. *Meets,* Second Monday. *President,* L. S. RIES, Oberlin College, Oberlin, Ohio. *Secretary,* C. E. PRICE, 6 N. Michigan Ave.

KANSAS CITY: *Organized,* 1917. *Headquarters,* Kansas City, Mo. *Meets,* Second Monday in Month. *President,* A. H. SLUSS, 827 Mississippi Ave., Lawrence, Kan. *Secretary,* GUSTAV NOTTBERG, 914 Campbell St.

MANITOBA: *Organized,* 1935. *Headquarters,* Winnipeg, Man. *Meets,* Fourth Thursday. *President,* D. F. MICHIE, 5B, 553 Wardlaw Ave.

MASSACHUSETTS: *Organized,* 1912. *Headquarters,* Boston, Mass. *Meets,* Second Tuesday in Month. *President,* JAMES HOLT, Massachusetts Institute of Technology, Cambridge, Mass. *Secretary,* JOHN TURNER, 285 Columbus Ave.

MICHIGAN: *Organized,* 1916. *Headquarters,* Detroit, Mich. *Meets,* First Monday after the 10th of the Month. *President,* F. J. FEELY, 950 Trombley Rd., Grosse Pointe Pk. *Secretary,* G. H. TUTTLE, 2000 Second Ave.

WESTERN MICHIGAN: *Organized,* 1931. *Headquarters,* Grand Rapids, Mich. *Meets,* Second Monday in Month. *President,* W. W. BRADFIELD, 901 Michigan Trust Bldg. *Secretary,* S. W. TODD, JR., 309 Paris, S. E.

MINNESOTA: *Organized,* 1918. *Headquarters,* Minneapolis, Minn. *Meets,* Second Monday in Month. *President,* R. E. BACKSTROM, Room 1981, First Natl. Bank Bldg., St. Paul, Minn. *Secretary,* F. C. WINTERER, 836 Juno St., St. Paul, Minn.

MONTREAL. *Organized,* 1936. *Headquarters,* Montreal, Que. *Meets,* Third Monday. *President,* G. L. WIGGS, University Tower. *Secretary,* C. W. JOHNSON, 630 Dorchester St., W.

NEW YORK: *Organized,* 1911. *Headquarters,* New York, N. Y. *Meets,* Third Monday in Month. *President,* W. E. HEIBEL, 11 West 42nd St. *Secretary,* T. W. REYNOLDS, 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

WESTERN NEW YORK: *Organized,* 1919. *Headquarters,* Buffalo, N. Y. *Meets,* Second Monday in Month. *President,* B. C. CANDEE, 19 Tremont Ave., Kenmore, N. Y. *Secretary,* W. R. HEATH, 119 Wingate Ave.

NORTHERN OHIO: *Organized,* 1916. *Headquarters,* Cleveland, O. *Meets,* Second Thursday in Month. *President,* PHILIP COHEN, 401 East Ohio Gas Bldg. *Secretary,* C. A. MCKEEMAN, Case School of Applied Science.

OKLAHOMA: *Organized,* 1935. *Headquarters,* Oklahoma City, Okla. *Meets,* Second Monday. *President,* E. F. DAWSON, University of Oklahoma, Norman, Okla. *Secretary,* E. W. GRAY, Box 1498, Oklahoma City, Okla.

ONTARIO: *Organized,* 1922. *Headquarters,* Toronto, Ont. *Meets,* First Monday in Month. *President,* G. A. PLAYFAIR, 97 Jarvis St. *Secretary,* H. R. ROTH, 57 Bloor St. W.

PACIFIC NORTHWEST: *Organized,* 1928. *Headquarters,* Seattle, Wash. *Meets,* Second Tuesday in Month. *President,* W. W. COX, 326 Columbia St. *Secretary,* M. N. MUSGRAVE, 314-9th Ave., N.

PHILADELPHIA: *Organized,* 1916. *Headquarters,* Philadelphia, Pa. *Meets,* Second Thursday in Month. *President,* L. P. HYNES, 240 Cherry St. *Secretary,* C. B. EASTMAN, 530 Brookview Lane, Brookline, Upper Darby, Pa.

PITTSBURGH: *Organized,* 1919. *Headquarters,* Pittsburgh, Pa. *Meets,* Second Monday in Month. *President,* M. L. CARR, P. O. Box 1646. *Secretary,* T. F. ROCKWELL, Carnegie Inst. Tech.

ST. LOUIS: *Organized,* 1918. *Headquarters,* St. Louis, Mo. *Meets,* First Tuesday in Month. *President,* G. W. F. MYERS, 3947 W. Pine Blvd. *Secretary,* D. J. FAGIN, 1344 Woodruff Ave.

SOUTHERN CALIFORNIA: *Organized,* 1930. *Headquarters,* Los Angeles, Calif. *Meets,* Second Tuesday in Month. *President,* E. H. KENDALL, 1978 S. Los Angeles St. *Secretary,* J. F. PARK, 1234 South Grand.

TEXAS: *Organized,* 1936. *Headquarters,* College Station, Texas. *President,* R. F. TAYLOR, 909 Banker's Mortgage Bldg., Houston, Tex. *Secretary,* W. H. BADGETT, Texas Engrg. Experiment Station, College Station, Tex.

WASHINGTON, D. C.: *Organized,* 1935. *Headquarters,* Washington, D. C. *Meets,* Second Wednesday in Month. *President,* L. OURUSOFF, 411 Tenth St., N. W. *Secretary,* L. F. NORDINE, Room 203, 734 Jackson Pl., N. W.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday in Month. *President,* J. H. VOLK, 1906 W. St. Paul Ave. *Secretary,* H. C. FRENTZEL, 3000 W. Montana St.

Manitoba Chapter Honors C. H. Turland

July 19, 1937. An informal dinner and meeting was held by Manitoba Chapter at the Fort Garry Hotel, Winnipeg, as a mark of appreciation to Secy. C. H. Turland, who is leaving Winnipeg for a new location on the Pacific Coast.

Thirteen members and guests were present, including Pres. D. F. Michie, J. B. Steele, E. J. Argue, William Glass, H. R. Eade, E. R. Miller, R. L. Kent, William Worton, Frank Thompson, L. G. Scott and C. H. Turland.

On behalf of Manitoba Chapter, J. B. Steele expressed the great appreciation of the members for the conscientious and energetic attention that Mr. Turland has given to the Chapter in his capacity as Secretary, and as a token of this appreciation presented Mr. Turland with a key chain.

Further appreciation and best wishes of the members for his new position on the Pacific Coast were expressed by Messrs. Argue, Thompson and Eade, bringing to a close a very enjoyable meeting.

Texas Chapter Elects New Officers

May 29, 1937. The final meeting of Texas Chapter for the season 1936-37 was held at College Station, Texas, beginning with a dinner at seven o'clock at the Aggieland Inn. Fourteen members and guests were present.

After the dinner, the group assembled in the Mechanical Engineering Shops and examined the laboratory set-up of the Texas Engineering Experiment Station testing propeller type fans. From there the Chapter members went to the newly air conditioned cotton fiber testing laboratories in the Petroleum Building, and then to the air conditioned experimental laboratory where the room used for comfort zone studies was inspected as well as the equipment for determining the thermal conductivity of building materials.

Chapter members and guests then met in the lecture room for a business and technical session when the minutes of the meeting held at Dallas were read and approved. The secretary, W. H. Badgett, reported that fourteen ballots had been cast in favor of the nominees selected by the Nominating Committee with no dissenting votes, and Pres. F. E. Giesecke announced

that the officers of the Chapter for 1937-38 would be:

President—R. F. Taylor
Vice-President—H. W. Skinner
Secretary—W. H. Badgett
Treasurer—I. E. Rowe
Board of Governors—J. A. Kiesling, M. L. Diver, C. L. Kribs, Jr.

After discussing the possibility of the A.S.H.V.E. holding a meeting in Texas, the Chapter voted to extend to the Society an invitation to come to Dallas in January, 1939.

Upon proper motion duly seconded, it was voted to amend the Chapter By-Laws as requested by the Council so that 10 members would be required to constitute a quorum instead of 5.

Following the business meeting, Prof. E. G. Smith gave an historical sketch of engineering acoustics, which was followed by a discussion by Prof. W. W. McCorkle, entitled, The Transmission of Sound in Ducts.

Western Michigan Chapter Holds Annual Meeting

May 25, 1937. The annual meeting of the Western Michigan Chapter was held at Walnut Hills Country Club, Lansing, Mich., when a large group was present to welcome Col. D. S. Boyden, Boston, Mass., president of the A.S.H.V.E.

Preceding the meeting, the annual golf tournament was played by 25 who registered, resulting in awards being made as follows: low gross—Henry Muir, Kalamazoo; low net—Tom Osberger, Grand Rapids; blind bogie—Bart Strudley, Lansing, and P. O. Wierenga, Grand Rapids; high gross—S. W. Todd, Jr., Grand Rapids; best dressed golfer—Emil Bauch, Grand Rapids; and most hopeless golfer—B. F. McLouth, Lansing.

A horseshoe pitching contest was also enjoyed by a number of the members.

At the business session, President Boyden was introduced and spoke in reference to the activities of the A.S.H.V.E., particularly in regard to research and codes.

The retiring president of Western Michigan Chapter, Prof. L. G. Miller, was enthusiastically applauded for his good work and the newly elected president, W. W. Bradfield, was appropriately inducted into office.

The entertainment consisted of an archery demonstration by Mr. Kibbe of the Wolverine Archery Co., Coldwater, Mich., and dancing by the Animated Steppers.

An excellent meeting was enjoyed and the Western Michigan Chapter is looking forward to a splendid year of constructive activity.

The newly elected officers are:

President—W. W. Bradfield
Vice-President—C. R. McConner
Secretary—S. W. Todd, Jr.
Treasurer—B. F. McLouth

Returns to London

J. K. M. Pryke recently elected to membership in the Society returned to his home in London on the Queen Mary after spending several months in the United States studying American heating, ventilating and air conditioning practice.

Between October 1936 and July 1937 Mr. Pryke visited the principal cities of the United States and Canada as a winner of the Robert Blair Fellowship 1936 awarded by the London County Council. These fellowships have been awarded annually since 1924 and two grants for 1936 covered the study of high speed trains in Germany and industrial and comfort air conditioning in the United States.

Mr. Pryke will submit a report of his studies and observations to the Consultative Committee on Engineering of the London County Council.

Research Chairman Takes New Station

Lt. Col. W. A. Danielson, chairman of the Committee on Research is now stationed at Fort Knox, Ky., in charge of the Quartermaster's Corps. Mail should be addressed to him at that point. Colonel Danielson reports much interesting work at this army post which stands guard over the largest stock of gold in the world. He invites any members of the A.S.H.V.E. who are in the vicinity to pay him a visit.

Speakers Bureau Established for Chapters

The Council of the Society authorized the establishment of a Speakers Bureau to supply local chapters with speakers of outstanding ability and Pres. D. S. Boyden appointed a Committee to prepare Rules and Regulations covering the operation of this Bureau. The Committee which recently completed its work consisted of John Howatt, Chicago, *Chairman*, C. T. Baker, Atlanta; Albert Buenger, Dayton; E. K. Campbell, Kansas City; R. H. Carpenter, New York; S. D. Peterson, Seattle, and O. J. Prentice, Chicago.

Through the efforts of this Committee regulations have been established for the organization and operation of this Bureau from the headquarters of the Society and 24 speakers have indicated their willingness to participate in the work during the 1937-38 season. Each Chapter will be entitled to call on the Speakers' Bureau for a maximum of three speakers and among the subjects to be presented are the following: Automatic Coal Firing, Economics of Air Conditioning, Effect of Air Conditions on Comfort and Health, Atmospheric Pollution, Air Conditioning by Parables, Physiologic and Health Factors of Air Conditioning, Hot Water Heating Practice, Forced Warm Air Heating, Air Conditioning Plant Operation, Heating, Ventilating and Air Conditioning of Paper Mills, Research—Past, Present and Future, District Heating, Pipe Line Refrigeration, Fundamentals of Refrigeration and Air Conditioning.

The 24 local chapters of the Society have been invited to use the service of the Speakers Bureau and many of them have already selected subjects for discussion during the next few months. The plan of furnishing speakers through this Bureau has met with the hearty approval of the Chapter Officers, and the Council has received many letters of commendation for this important contribution to the technical discussions at local chapter meetings.

Certificate Course At New York University

A course in heating, ventilating and air conditioning at the College of Engineering, New York University, will be given in the evening division commencing September 14. This two-year course leads to a certificate of proficiency and is intended to assist those now employed in the design, erection, operation, or manufacture of equipment, those who wish to enter these specialized fields of engineering, and those who desire to review or amplify their knowledge of heating, ventilating and air conditioning.

Celotex Appoints

An announcement by Harold Knapp, vice president and general sales manager, The Celotex Corporation, Chicago, Ill., states that George E. Swenson becomes manager of sales of the Metuchen Division and Paul D. Close is appointed assistant manager with headquarters in New York City. For several years Mr. Close has been chief engineer of the Industrial Uses Division.

Annual Reports of Local Chapters

Koehne Photo

L. S. Ries

C. E. Price

Illinois Chapter

The Officers elected for the 1937-38 season are:

President—L. S. Ries
Vice-President—S. I. Rottmayer
Secretary—C. E. Price
Treasurer—J. R. Vernon
Board of Governors—M. W. Bishop, Tom Brown, and E. M. Mittendorff.
Legislative Committee—R. E. Hattis, Chairman, John Howatt and J. H. Milliken.
Membership Committee—E. M. Mittendorff, Chairman, C. M. Baum-Gardner, B. A. Broom, J. T. Machen. G. G. Turner.
Meetings and Publications Committees—Tom Brown, Chairman, S. I. Rottmayer, H. F. Reid, V. L. Sherman, and N. W. Swanson.
Finance Committee—J. R. Vernon, Chairman, J. C. Matchett, E. W. Rietz, O. W. Armspach and M. W. Bishop.

The annual report of the Board of Governors of the Illinois Chapter covering activities for 1936-37 is submitted by Secretary C. E. Price. The total membership in May 1937 was 147 members of all grades and the average attendance at meetings was 116.

At the Annual Meeting on May 11, 1936, the following Officers and members of the Board of Governors were elected:

President—J. J. Hayes
Vice-President—L. S. Ries
Secretary—C. E. Price
Treasurer—J. Rex Vernon
Board of Governors—O. W. Armspach, C. E. Crone, Jr., S. I. Rottmayer and J. H. Milliken

The following Committees were appointed by President Hayes to carry on the regular business of the Chapter:

Membership Committee—Tom Brown, Chairman, M. W. Bishop, J. R. Borling, C. E. Crone, E. M. Mittendorf, F. I. Raymond and E. W. Rietz.
Meetings and Publication Committee—L. S. Ries, Chairman, C. F. Hayden, P. J. Marschall, J. H. Milliken, C. E. Price and S. I. Rottmayer.
Finance Committee—J. R. Vernon, Chairman, B. L. Casey, J. P. Fleming and E. N. McDonnell.
Legislative Committee—J. H. O'Brien (1937), R. E. Hattis (1938) and John Howatt (1939).

In addition to these regular Chapter Committees a special committee to investigate the water supply and disposal problem in connection with conditioning in Chicago's central business district was appointed by President Hayes immediately after the first meeting of the year. The Committee consisted of O. W. Armspach, Chairman, H. M. Hart, John Howatt, G. W. Hubbard and Louis Narowetz.

The regular Chapter meetings with the speakers and subjects of their talks were as follows:

October 12, 1936. Loran D. Gayton, acting city engineer of Chicago spoke on Air Conditioning.

November 9, 1936. The speaker of the evening was G. L. Larson, President of the A.S.H.V.E. who spoke on The Economics of Good Construction as Related to Residence Heating.

December 14, 1936. The subject Cooling Requirements for Summer Comfort Air Conditioning was given by John Howatt, Board of Education.

January 11, 1937. H. J. Rogan, assistant chief engineer, The Estate of Marshall Field gave a talk on Building Owning Industry Views Air Conditioning Industry.

February 8, 1937. This meeting was a Dinner and Dance for the Ladies. The speaker of the evening was Dr. Watson Boyes, Secretary of Oriental Institute, University of Chicago who related Engineering Feats in the Ancient Near East.

March 8, 1937. A joint meeting with the Milwaukee members of the Society was held and R. E. Hattis gave an interesting talk on Air Conditioning the Windowless Building of National Aluminate Corp.

April 12, 1937. Clarkson Coil Steam Generating was the title of the subject given by Alick Clarkson, consulting engineer.

G. L. Wiggs

C. W. Johnson

Montreal Chapter

The following Officers and Committees are in charge of the Montreal Chapter which has a membership of 55:

President—G. L. Wiggs.
Vice-President—F. J. Friedman.
Secretary—C. W. Johnson.
Treasurer—F. G. Phipps.
Board of Governors—A. B. Darling, W. G. Dewar, L. Garneau, W. U. Hughes and J. Linton.
Membership Committee—A. B. Darling, Chairman, H. Gendron, A. Lamontagne, A. G. Marshall and L. Martin.
Attendance Committee—L. Garneau, Chairman, F. R. Barnsley and T. Worthington.
Publicity Committee—G. L. Ballantyne, Chairman, L. H. Laffoley and J. A. M. Robertson.

A summary of the meetings for the 1936-37 season is given herewith.

October 14, 1936. The meeting was held at the Windsor Hotel with 70 members and guests in attendance. Prof. G. L. Larson, President of A.S.H.V.E. and head of the Dept. of Mech. Engineering, University of Wisconsin gave an illustrated talk on The Economics of Good Construction as Related to Residence Heating. A. V. Hutchinson, Secretary of the Society spoke on Research Activities and Program of the Society.

November 9, 1936. At this meeting there were 80 members and guests present, which was the largest attendance for the season. The speaker was W. L. Fleisher, consulting engineer whose talk was on Evaporative Cooling for Comfort.

December 14, 1936. A paper was presented by Cyril Tasker of the Ontario Research Foundation on Cooling Requirements for Summer Comfort Air Conditioning, before an audience of 40 members and guests.

January 11, 1937. With an attendance of 44 members and guests John James, Technical Assistant of the Society gave a talk on A Review of Residence Air Conditioning Research.

February 15, 1937. The speaker of this meeting was E. H. Gurney, First Vice-President of the Society, who gave a very interesting talk on Heating and Air Conditioning Observations in South Africa. Attendance was 39.

March 15, 1937. A. W. Moulder, Vice-President of Grinnell Co. gave a lecture on Forced Hot Water Heating which brought out the history of the development of hot water heating control.

engineering in detail back of design of hot water systems and basic fundamentals. The attendance was 38 members and guests.

April 19, 1937. With an audience of 40 members and guests, John Everetts Jr. of Air and Refrigeration Corp. gave an interesting and informative talk on Refrigeration as Pertaining to Steam Jet and Water Vapor Machines.

May 19, 1937. This meeting held at the Windsor Hotel, was a General Business Session with Election of Officers.

June 16, 1937. A combined Golf and Dinner Meeting was held at The Whitlock Golf and Country Club with 23 members and guests in attendance.

W. E. Heibel

New York Chapter

The following members were elected to office for the season of 1937-1938:

President—Walter E. Heibel.
Vice-President—Dexter Purinton.
Secretary—T. W. Reynolds.
Treasurer—W. M. Heebner.
Board of Governors—G. E. Olsen, W. J. Osborn, Howard G. Meinke.

The annual report of the New York Chapter as prepared by Secretary T. W. Reynolds for the period from October 1936 to May 1937 gives an average attendance at meetings of 77 and a membership of 168 at the close of the season.

October 19, 1936. The first meeting of the season was devoted to a discussion of air conditioning as applied to fever therapy. W. L. Fleisher, Council Member and member of A.S.H.V.E. Committee on Research, outlined the opportunities for cooperation between the engineer and physician in regard to the use of high humidities and temperatures for the treatment of diseases. Dr. M. B. Ferderber gave a preliminary report on the results of such studies as conducted at St. Francis Hospital, Pittsburgh. F. C. Houghten, Director, A.S.H.V.E. Research Laboratory, reviewed the development of the fever box and W. J. McConnell, M.D., Assistant Medical Director, Metropolitan Life Insurance Co. the final speaker of the evening, cited some interesting applications of conditioned air to the application of fever therapy.

November 16, 1936. The Economics of Good Construction in Heating and Air Conditioning was the subject of this meeting. The speaker, G. L. Larson, President of A.S.H.V.E. and Chairman, Department of Mechanical Engineering, University of Wisconsin, discussed the problems in heating and air conditioning confronting heating engineers, builders and home owners. He gave a tabulation and analysis made in accordance with modern engineering practice with the primary cost of insulation, weatherstripping, storm windows and resulting fuel saving graphically shown. An illustrated discussion on the cost of humidification was also given by Prof. Larson.

December 21, 1936. This was an evening given over to the annual card party, a social event which brought out a large attendance of members and their wives.

January 18, 1937. Humidity, Its Control and Measurement, was the feature of this meeting. The speakers were Ralph R. Chappell and Kenneth E. Whitney of Julien P. Friez & Sons, Inc.

February 15, 1937. Robert Percival Moses, consulting engineer, addressed the meeting on The Return of the Private Generating Plant in Large New York City Buildings. Mr. Moses presented figures on the cost of installation, maintenance and depreciation, and the operating economy of private plants in comparison with electricity purchased from public utilities, together with engineering data on boilers, engines, generators, and other mechanical equipment.

March 15, 1937. Some new facts on Radiant Heating and Cooling were presented. The latest developments in experimental theory and practice in domestic and commercial installations in Europe and this country were covered by illustrated views and motion pictures. A review of the work done along theoretical and experimental lines was discussed by Dexter Purinton, Engineering Vice-President, Mahony-Troast Construction Co. The extensions of the theory to include installations of a practical nature were reviewed by Orion O. Oaks, Chief Engineer, American Radiator Co., Leon Munier, Wolff & Munier, Inc. and G. W. Penney of Westinghouse Electric and Mfg. Co.

April 19, 1937. Walter L. Fleisher, Consulting and Advisory Engineer, spoke on the subject of Air Conditioning of Existing Large Buildings. Mr. Fleisher pointed out that the air conditioning of large buildings presents a serious problem in finance and difficult problems of an engineering nature, also that an ever increasing ratio in the number of buildings using air conditioning equipment has brought about keen competition in related business such as Restaurants, Theaters and Department Stores and from this a full realization that air conditioning must be introduced into such places or they fall by the wayside. The cost of doing such work, together with the economics and engineering problems involved were ably discussed by Mr. Fleisher from his fund of information and experiences along these lines. His remarks on the subject of air conditioning were therefore of great interest to all who attended.

May 17, 1937. The annual dinner, smoker and entertainment under the direction of Harry W. Fiedler and his committee, took place at the Building Trades Club. This was a most successful affair with the usual high attendance, a fine dinner, good music, and entertainment which was enjoyed by all.

L. P. Hynes

C. B. Eastman

Philadelphia Chapter

C. B. Eastman, secretary of the Chapter, gives a resumé of the meetings held during the 1936-37 season and reports that the Chapter membership totals 116. The following are the Officers and Committees functioning during the year:

President—L. P. Hynes.
Vice-President—H. H. Erickson.
Secretary—C. B. Eastman.
Treasurer—R. F. Hunger.
Board of Governors—A. E. Kriebel, H. H. Mather and W. F. Smith.
Meetings Committee—H. H. Erickson, Chairman, E. A. Dambly, P. L. Davidson, L. E. Moody and H. F. Rettew.
Membership Committee—H. H. Mather, Chairman, H. E. Augenbaugh, C. G. Binder, A. G. Dome, H. B. Hedges, J. M. Jopson, M. G. Kershaw, Morris Sheffler, L. A. Tucker and C. W. Whitney.
Finance Committee—R. F. Hunger, Chairman, H. G. Black, Edwin Elliot, W. A. Bornemann and J. H. Hucker.
Special Advisory Committee—H. P. Gant, Chairman, F. D. Mensing and W. F. Smith.
Legislative Committee—M. F. Blankin. Chairman, W. P. Culbert and C. S. Leopold.
Editor of the Year Book—George W. Barr.
Year Book Advertising Committee—P. L. Pryibil, Chairman, Robert Arnold, Lucien Buck, A. C. Caldwell, H. P. Dever, A. E. Kriebel and S. E. Plewes.

Dinner Committee—A. H. MacDade, Chairman, A. M. Morris and H. M. Patrick.

On to Swampscott Committee—H. H. Erickson, Chairman, and A. E. Kriebel.

Golf Committee—W. P. Culbert, Chairman, and R. C. Bolsinger.

An outline of the activities during the year is given as follows:

October 8, 1936. The meeting was held at the Engineers Club and the speaker was B. S. Weaver of General Electric Co. who delivered a paper entitled, Fractional Horsepower Motors—Their Selection and Application to the Heating, Ventilating and Air Conditioning Industry. The paper was accompanied by slides. Attendance, 50 members and guests.

November 12, 1936. The President of the Society, Prof. G. L. Larson spoke on The Economics of Good Construction as Related to Residence Heating which was illustrated by slides. There were 72 members and guests in attendance.

December 10, 1936. The speaker for this evening, Charles S. Leopold, talked on The Various Types of Air Conditioning Systems, to an audience of 125 members and guests.

January 14, 1937. This was a "play" meeting at the Hudson Recreation Centre, with a Dutch lunch and bowling for the 60 members and guests, who attended.

February 11, 1937. A. E. Stacey, Jr. delivered a paper to 47 members and guests entitled, Studies Made on Corrosion.

March 11, 1937. Surface Cooling was the subject on which W. R. Heath, Asst. Chief Engineer of the Buffalo Forge Co. spoke to 40 persons in attendance.

April 8, 1937. Two speakers were on the program, namely, Paul Young of Johns-Manville Corp. who presented a paper entitled Sound Insulation and Isolation, and Arthur M. Simpson, general sales manager of Van Kannel Revolving Door Co. who presented the paper, Infiltration Through Doorways. Both papers were illustrated by slides and film. The annual election of Officers was also held. Attendance was 43.

May 13, 1937. The Annual Golf Outing was held at the Aronomink Golf Club, Newton Square, Pa., dinner being served to 87 members and guests in the grill room of the club.

M. L. Carr

T. F. Rockwell

Pittsburgh Chapter

The annual report of the Pittsburgh Chapter, prepared by T. F. Rockwell shows 55 members on the roll, a net gain of 10 during the last year.

The officers and committees of the Pittsburgh Chapter are as follows:

President—M. L. Carr
Vice-President—J. F. Collins
Treasurer—J. E. McLean
Secretary—T. F. Rockwell
Board of Governors—R. J. J. Tennant, Chairman, Arthur MacGonagle and R. A. Miller
Membership Committee—P. A. Edwards, Chairman, L. S. Maehling, F. C. McIntosh, E. H. Reismeyer, Jr. and R. B. Stanger.
Program Committee—J. F. Collins, Jr., Chairman, R. H. Heilman, R. A. Miller, John Proie and E. C. Smyers.
Air Conditioning Code—G. S. McEllroy, Chairman, P. A. Edwards, F. C. Houghten and G. G. Waters.
Degree Day Committee—F. A. Gunther, Chairman, J. F. Collins, Jr., F. C. Houghten.
Social Committee—E. S. Tower, Chairman, V. A. Reed, Chas. Sonneborn, P. A. Strauch and R. J. J. Tennant.
Atmospheric Pollution Committee—M. I. Dorfan, Chairman, T. G. Estep, J. E. Frazier and J. H. Holden.

The following is a resumé of the meetings during the past season:

October 12, 1936. An inspection trip, to the Stanwix Plant—Allegheny County Steam Heating Co. with W. W. Stevenson acting as guide, was attended by 15 members and guests.

November 16, 1936. F. C. Houghten, Director of Research Laboratory A.S.H.V.E. spoke in the U. S. Bureau of Mines Auditorium to 20 members and guests on Summer Cooling Requirements.

December 14, 1936. Thirty-five members and guests heard an interesting talk on A Trip Through Africa by George Evans.

January 11, 1937. There were two speakers on this program, namely, J. F. Collins, Jr. and H. A. Cannon who spoke on Air Conditioning an Office Building. The attendance was 42 members and guests.

February 8, 1937. The following gentlemen took part in the evening's program, namely, H. A. Cannon, F. C. McIntosh, Leighton Orr and M. W. Lightcap. The subject was Air Conditioning an Office Building. 48 members and guests were in attendance.

March 8, 1937. A discussion of the subject of Air Conditioning an Office Building was participated in by the following men, J. L. McCullough, E. C. Smyers, F. C. McIntosh, P. A. Edwards and V. W. Hunter. Attendance—48.

April 19, 1937. An audience of 47 members and guests heard E. H. Gurney, first vice president of the Society speak on An Engineer Goes Sightseeing.

May 10, 1937. The subject Atmospheric Pollution was discussed by M. I. Dorfan, H. B. Meller, Dr. Frank Holden and H. R. Loxterman before an audience of 37 members and guests.

E. H. Kendall

Southern California Chapter

The Officers of the Chapter are as follows:

President—E. H. Kendall.
Vice-President—H. M. Hendrickson.
Secretary—J. Frank Park.
Treasurer—A. W. Cooper.
Board of Governors—W. D. Fabling, Leo Hungerford, O. W. Ott, and P. C. Scofield.

The following resumé of the meetings during the past year has been contributed by J. Frank Park, secretary of the Chapter:

October 13, 1936. O. W. Ott, chairman of the Chapter's Committee on Research reported progress in formulating a definite program and outlined a plan of sponsoring research in cooling towers and surface coolers at the University of California. A short talk outlining The Progress of Air Conditioning in the Last Ten Years was given by A. J. Hess. Forty-six members and guests were in attendance.

November 10, 1936. Dr. B. M. Woods, of the Department of Mechanical Engineering of the University of California spoke to 52 members and guests on important research problems in air conditioning. Motions were made and passed requesting financial and other aid from the parent Society and setting up a committee to handle the matter.

December 8, 1936. After a report by O. W. Ott on the progress of the Committee on Research, Leo Hungerford spoke on the subject, Willis H. Carrier's Contribution to the Air Conditioning Industry, in honor of the 25th Anniversary of Mr.

Carrier's presentation of the Rational Psychrometric formula. There were 25 members and guests in the audience.

January 12, 1937. J. K. Horton the speaker of the evening presented the Robinson-Patman Act and Its Relationship to the Contracting and Sub-contracting Business, to 37 members and guests.

February 16, 1937. A joint meeting of this Chapter and the California Chapter of the *American Society of Refrigerating Engineers* was held. H. M. Hendrickson reported on the 43rd Annual Meeting of the A.S.H.V.E. and a supplementary report was made by Mr. Ott advising that the parent Society would probably approve the proposed research to be sponsored in conjunction with the University of California. The principal speaker of the evening was P. H. Thompson, chief engineer of Alco Valve Co. who spoke on Refrigerant Control Valves, which was sponsored by the *A.S.R.E.*

March 3, 1937. Another joint meeting with the California Chapter of the *A.S.R.E.* was held and David L. Fiske, Secretary of the *A.S.R.E.* gave an interesting talk on How Not to Run a Business, which was followed by considerable discussion of his theories. 108 members and guests of the two Societies were present.

April 20, 1937. This was President's Night at which the *A.S.R.E.* members joined in honoring Pres. D. S. Boyden, who spoke on The Economical Use of Purchased Steam. He also discussed the general conditions of the A.S.H.V.E. The attendance of members and guests was 91.

May 11, 1937. The largest attendance for the year was at this meeting which totaled 211 members and guests. The principal speaker was Dr. B. M. Woods, of the Mechanical Engineering Department of the University of California, who spoke on The Progress in Air Conditioning Studies. O. W. Ott then reported approval of the proposed research program and that the Society had granted $600 toward the cost of the program. The San Francisco group signified their intention of subscribing $300 toward the work. He reported that the local Chapter had their amount pledged. R. L. Gifford was presented a Life Membership Certificate by O. W. Ott.

The total membership on the Chapter rolls is 37.

R. F. Taylor

W. H. Badgett

Texas Chapter

The Texas Chapter of the American Society of Heating and Ventilating Engineers was granted a charter by the Council on November 11, 1936, and the first meeting of the new chapter was held at College Station, Texas on December 19, 1936 at which time the following Officers were elected:

President—F. E. Giesecke.
Vice-President—R. F. Taylor.
Secretary—W. H. Badgett.
Treasurer—I. E. Rowe.
Board of Governors—M. L. Diver, J. A. Kiesling and C. L. Kribbs.

A brief review of the meetings held is given by W. H. Badgett, Secretary of the Chapter.

December 19, 1936. A resumé of Research Relating to the Air Conditioning of Railway Cars was given by Prof. W. E. Long. The next speaker on the program was Prof. E. G. Smith, who spoke on Sound and Air Conditioning. The close of the meeting was a report on research of the Texas Engineering Experiment Station Relating to Air Conditioning by Prof. F. E. Giesecke.

March 20, 1937. A dinner at the Rice Hotel was attended by 19 members and guests. R. M. Spencer, Powers Regulator Co. discussed the subject, Automatic Controls in Air Conditioning and C. A. McKinney of the United Gas Co. presented a paper entitled The Uses of Gas in Air Conditioning.

April 15, 1937. The Charter for the Texas Chapter was presented by Col. D. S. Boyden, President of the Society, who spoke briefly on the high points in the Society's history and accomplishments, and outlined research work in air conditioning now being undertaken by the Society. W. H. Carrier, chairman of the board of Carrier Corp. also spoke on the history of air conditioning and described the air conditioning of the deep gold mines in South Africa. Col. Boyden then spoke on the subject of Economic Uses of Purchased Steam.

May 29, 1937. Fourteen members and guests had dinner at the Aggieland Inn, after which they examined the Texas Engineering Experiment Station's Laboratory, set-up for testing propeller type fans, the air conditioned laboratory for comfort zone studies and the equipment for determining the coefficient of thermal conductivity of building materials. The group then visited the air conditioned cotton fiber testing laboratory of the USDA in the Petroleum Building.

The Officers elected to serve in 1937-38 are:

President—R. F. Taylor.
Vice-President—H. W. Skinner.
Secretary—W. H. Badgett.
Treasurer—I. E. Rowe.
Board of Governors—J. A. Kiesling, M. L. Diver, C. L. Kribbs, Jr.

L. S. Ourusoff

Washington, D. C. Chapter

The general activities of the Washington Chapter indicate a continuing interest in the work carried on during the 1936-37 season with a membership of 40. Attendance at meetings has been exceptionally good and a number of visitors at each meeting illustrates the local interest in Chapter work. The Officers and Committees serving for the year beginning May 1937 are:

President—L. S. Ourusoff.
Vice-President—S. P. Eagleton.
Secretary—L. F. Nordine.
Treasurer—Glegge Thomas.
Board of Governors—M. D. Kiczales, W. E. Kingswell and T. H. Urdahl.
Membership Committee—E. H. Lloyd, Chairman, D. M. Robinson, Assistant Chairman, F. M. Grimes, A. E. Stack, F. M. Thuney.
Publicity Committee—T. H. Urdahl, Chairman, E. V. Fineran, Assistant Chairman, W. R. Gifford, S. L. Gregg and J. B. Hewett.
Program Committee—L. S. Ourusoff, Chairman, W. H. Loving, Assistant Chairman, G. S. Frankel, L. F. Nordine and R. D. Terhune.
Attendance Committee—L. B. Nye, Jr., Chairman, W. Dewey Marshall, Assistant Chairman, H. R. Iverson, C. S. Stock and E. E. Sutter.
Public Relations Committee—E. J. Febrey, chairman, E. H. Lloyd, Assistant Chairman, E. S. DeWitt, L. S. Ourusoff and T. H. Urdahl.
Nominating Committee—R. H. Feltwell, Chairman, J. B. Hewett and C. S. Stock.

A review of the meetings held during the past season is given herewith by L. F. Nordine, Secretary of the Chapter.

October 14, 1936. With 60 members and guests in attendance, Col. W. A. Danielson, President of the Chapter, described his trip through Alaska and told of climatic conditions, building con-

struction and methods of heating and ventilating which he had observed. J. B. Hewett, General Electric Co., Washington, D. C., described the design and installation of the air conditioning system of Hotel Harrington and after a discussion of the paper, a tour of inspection was made.

November 11, 1936. An illustrated talk on the principles of design of a gas heating unit was given by G. F. Miller, C. A. Dunham Co., Washington, D. C., to an audience of 50 members and guests.

December 9, 1936. Greenhouse heating was the subject presented by L. F. Nordine with 35 members and guests in attendance.

January 12, 1937. A discussion of comfort conditions based on Research Laboratory investigations was given by F. C. Houghten, Director of the A.S.H.V.E. Research Laboratory and 100 members and guests enjoyed this illustrated talk.

February 10, 1937. R. A. Miller, Pittsburgh Plate Glass Co. gave an interesting paper on Glass Blocks, Their Insulation Value and Light Transmission, before an audience of 60 members and guests.

March 10, 1937. The subject of Forced Hot Water Heating was discussed by two speakers, A. W. Moulder, Vice-President of Grinnell Co. and C. E. Pullum, Vice-President of Bell and Gossett Co. This discussion was enjoyed by 60 members and guests.

April 9, 1937. The Chapter was host to the Officers and Council of the Society on this occasion and the speakers were D. S. Boyden, President of the Society, E. H. Gurney, First Vice-President and Col. W. A. Danielson. The Officers and Council in attendance were: D. S. Boyden, President, E. H. Gurney, First Vice-President, A. J. Offner, Treasurer, A. V. Hutchinson, Secretary and J. J. Aeberly, M. C. Beman, R. C. Bolsinger, S. H. Downs, W. L. Fleisher, F. E. Giesecke, G. L. Larson, W. A. Russell and W. E. Stark.

Among the other 65 members and guests were F. C. Houghten, J. D. Cassell and M. F. Blankin.

May 12, 1937. The election of Officers was held during the meeting and the featured speaker was Herbert Peacock, Carrier Corp., Washington, D. C. who spoke on Refrigeration by Centrifugal Compressors. Thirty-nine members and guests were in attendance.

E. F. Dawson

E. W. Gray

Oklahoma Chapter

The Oklahoma Chapter membership numbers 15 on the roll and the following are the Officers:

President—E. F. Dawson
Vice-President—R. E. Paulmg
Secy.-Treas.—E. W. Gray
Board of Governors—R. G. Dolan, Edwin Jones and S. L. Rolland

The following is a synopsis of the meetings held during the past year:

October 12, 1936. Attendance—28. Speaker—Prof. F. E. Giesecke, of Texas A. & M. College. Subject—History and Principles of Air Conditioning and Current Research.

November 11, 1936. Attendance—20. Speaker—C. A. Thinn, chief engineer of the C. A. Dunham Co. Subject—Differential System of Heating.

December 14, 1936. Attendance—18. Speaker—E. F. Dawson, vice-president of the Chapter. Subject—Cooling Requirements for Summer Comfort Air Conditioning.

January 11, 1937. Attendance—25. Speaker—M. A. Disney of the Disney-Leffel Co. Subject—Air Distribution.

February 8, 1937. Attendance—35. Speaker—A. B. Newton, Minneapolis Honeywell Regulator Co. Subject—Application of Controls to Modern Air Conditioning Equipment.

March 8, 1937. Attendance—20. Speaker—E. F. Dawson, vice-president of the Chapter. Subject—Refrigeration Fundamentals and Practical Application in Air Conditioning.

April 14, 1937. A dinner in honor of President and Mrs. D. S. Boyden was held at the Oklahoma Club followed by the regular meeting with 20 members and guests present. President Boyden outlined the history and aims of the Society followed by an illustrated talk on the characteristics of district steam heating business.

World's Fair Administration Building Completed

A'most a year ago when the New York World's Fair 1939 was still a paper project and the site one tremendous cloud of dust, Fair engineers picked August 15, 1937, as the date on which the Administration Building—not even designed at the time—would stand completed and ready for occupancy.

The engineers missed their calculation by 2 days, for on Aug. 13 the ceremonies marking the completion of the $900,000 structure were held. Two days ahead of the schedule and not only is the building ready for 600 Fair employees, but the four-acre plot in which it stands, adjacent to the Long Island Railroad bridge over Grand Central Parkway, has been completely landscaped with trees, shrubs, flowers and green lawns.

To signalize the finish in record time of the Fair's first major construction job Grover Whalen, president, raised the flags of nation and Fair on flagpoles at either side of the main entrance to the two-story structure. He then pulled a cord and a large rectangle of cloth fell away from the facade revealing the sculptured figure of a woman semi-draped, rising above an island city and lifting with upraised hands the veil that shrouds the World of Tomorrow.

Accompanied by Fair officials, reporters and photographers, Mr. Whalen next entered the building for a tour of inspection of its 115,000 sq ft of floor space.

Design of the building, which covers an acre and one-third of ground, was begun Oct. 21, 1936, by the four collaborating architectural firms of Harvey Stevenson & Eastman Studds, John A. Thompson & Gerald Holmes, Edgar I. Williams and Kimball & Husted. Architectural plans were finished Jan. 1, 1937. By that time work preliminary to construction already was under way.

On April 27 the cornerstone was laid, and three weeks later the last steel rivet had been hammered home. During the following month the building was sheathed in wood and coated with stucco. Plumbing, heating, air-conditioning and electrical systems were installed, and an army of carpenters, plasterers and painters descended on the interior. Until the last minute the structure swarmed with workmen busy on finishing touches, but Fair engineers had everything in perfect order for the opening ceremonies.

R. B. Hayward Dies Suddenly

The sudden passing of Ralph B. Hayward, well known heating and ventilating engineer of Chicago, while on vacation in Lockport, N. Y., was a great shock to his many friends and business associates. While on a lake cruise enroute to his old home, he suffered a heart attack at Buffalo and was taken to Lockport hospital where he died on Tuesday, August 24, at the age of 63 years.

Ralph Baker Hayward was born in Lewiston, N. Y., September 10, 1874. After attending the public schools in Lewiston

he went through the high school in Lockport. Then he entered Cornell University, graduating from the engineering course in 1899, with a degree of M.E. He worked as a machinist for Southern Railway in Atlanta, Ga., and then joined the engineering staff of Buffalo Forge Co. From 1902 to 1906 he was sales engineer for B. F. Sturtevant Co. in Pittsburgh and for the next two years served as Chicago Branch Manager of Buffalo Forge Co. The Mellish-Hayward Co. was then organized and Mr. Hayward was its Treasurer. Later, the company became the R. B. Hayward Co. and it handled some of the largest sheet metal work for the heating, ventilating and air conditioning of some of Chicago's most important buildings.

Mr. Hayward was always interested in civic affairs, serving the village as trustee of LaGrange, Ill., from 1921 to 1925 and this was followed by two years as president of the board. He was likewise interested in the arts, especially music and drama, and for several years had served as president of the LaGrange Players, assisting them in many ways in the productions.

Mr. Hayward was a member of the Presbyterian church and the Masonic Lodge. He became a member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in 1909 and served on many technical committees. He was an active member of Illinois Chapter and served on its Board of Governors. He belonged to the University Club, the LaGrange Country Club and a local reading club. His recreations were golf and motoring, but probably his principal interest was in his home and beautiful yard at 201 South Stone Ave., LaGrange.

Mr. Hayward was married in 1901 to Mary E. McMaster at Lockport, who survives with two daughters, Mrs. Charles (Jeanette) Yeatman of Los Angeles and Mrs. Reuben (Rebecca) Frodin of San Francisco; one brother, W. J. of Sioux City, Ia.; and three sisters, Mrs. L. G. Merritt of Lockport, Mrs. W. S. Hovey of Deland, Fla., and Mrs. J. J. Ricks of Plandome, L. I.

Funeral services were held in Lockport Wednesday afternoon, followed by interment in the old family cemetery.

The Officers and Council of the Society regret the passing of a loyal and valued member, an interested and active worker, and extend their sincere sympathy to his family.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 32 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by September 15, 1937, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ANAYA, MARVIN, Mech. Design, Bureau of Engineering, City Hall, San Francisco, Calif.	J. I. Krueger R. A. Hudson	W. E. Leland L. H. Cochran
DAUBER, O. W., Consulting Engr., Chicago, Ill.	John Howatt H. R. Linn	E. P. Heckel B. A. Broom
DAVEY, G. I., Consulting Engr., Haskins & Davey, Sydney, Australia.	A. G. Gutteridge (*I. C. E.*) Gerald Haskins (*I. C. E.*)	G. R. Goffin (*I. C. E.*) K. P. Inglis (*Non-Member*)
DONELSON, W. N., Htg. Engr., T. J. Conners, Inc., Cincinnati, Ohio.	L. F. Hudepohl R. E. Hust	A. W. Edwards E. B. Royer
ELBERT, B. F., Sales Engr., Airtemp Div., Sidles Co., Lincoln, Nebr.	F. E. Prawl M. J. Stevenson	I. C. Baker L. L. Daubert
ELLIOTT, IRWIN, Chief Engr., Universal Oven Co., New York, N. Y.	L. A. Rusch (*Non-Member*) G. E. Ramsden (*Non-Member*)	N. Cohan (*Non-Member*) A. W. Drake (*Non-Member*)
FANSLER, P. E., Editor, Air Conditioning Combined with Oil Heat, New York, N. Y. (*Reinstatement*)	J. D. Hoffman C. H. B. Hotchkiss	Homer Addams Thornton Lewis
FEIRN, W. H., Htg. & Vtg. Engr., C. A. Hopper Co., Madison, Wis.	G. L. Larson D. W. Nelson	J. C. White Ernest Szekely
FERRARINI, JOSEPH, Testing Engr., Washington Gas Light Co., Washington, D. C.	A. E. Stack P. H. Loughran, Jr.	W. H. Loving E. V. Fineran
FOERSTNER, G. C., Mgr. Air Cond. Dept., Amana Society, Amana, Iowa.	A. G. Matousek L. J. Du Bois	H. A. Ware (*Non-Member*) J. M. Musser (*Non-Member*)
FOOTE, A. G., Dist. Engr., Frigidaire Div., Oakland, Calif.	C. S. Ames N. H. Peterson	G. J. Cummings C. E. Bentley
GAYNOR, JAMES, Mech, Engr., G. M. Simonson, San Francisco, Calif.	A. J. Bouey L. H. Cochran	G. J. Cummings C. S. Ames
GOUNDIE, J. K., Sales Engr., Fritch Coal Co., Bethlehem, Pa.	Pensyl Mawby C. B. Korn	R. H. Snoeberger (*Non-Member*) P. T. Frankenfield (*Non-Member*)

CANDIDATES	*Proposers*	REFERENCES *Seconders*
HALLECK, L. P., Vice Pres. & Sales Mgr., The Allen Corp., Detroit, Mich.	W. A. Russell W. G. Boales	R. F. Connell J. F. McIntire
HUBBARD, N. B., Consulting Engr., Hubbard & Wagschal, Detroit, Mich. (*Reinstatement*)	W. G. Boales E. H. Clark	W. A. Russell A. V. Hutchinson
JACKES, H. D., Sales Mgr., Aerofin Corp., Bloomfield, N. J. (*Reinstatement*)	H. L. Baker, Jr. John Everetts, Jr.	Arthur Ritter W. E. Heibel
KENNEY, T. W., Sales Mgr., Jas. P. Marsh Corp., New York, N. Y.	E. J. Ritchie W. W. Timmis	C. J. Kelly F. B. Campbell
MCCARTHY, J. J., Engr., Providence School Dept., Providence, R. I.	E. M. Fenner E. W. Gibbs	J. D. McLaughlin C. W. Kimball
MCCLAIN, C. H., Htg. & Air Cond. Engr., Upper Darby Plbg. & Htg. Co., Inc., Upper Darby, Pa.	C. B. Eastman R. C. Bolsinger	E. N. Black, 3rd A. E. Kriebel
NOTTBERG, HENRY, JR., Engr., U. S. Engineering Co., Kansas City, Mo.	Carl Clegg A. H. Sluss	C. A. Weiss Gustav Nottberg
RANDALL, R. D., Sales Repr., D. T. Randall & Co., Detroit, Mich. (*Reinstatement*)	G. H. Giguere E. H. Clark	W. G. Boales G. D. Winans
ROGERS, R. C., House Htg. Engr., Community Natural Gas Co., Bryan, Texas.	W. H. Badgett F. E. Giesecke	G. M. Hines C. E. Griesser
SAMUELS, SIDNEY, Pres., Sidney Samuels, Inc., New York, N. Y. (*Reinstatement*)	J. K. Peacock N. P. Fenner	C. V. Haynes E. N. Sanbern
SCOTT, A. F. H., Asst. to Pres., Anthracite Industries, Inc., New York, N. Y.	G. L. Larson L. N. Hunter	A. J. Johnson Homer Addams
SHAPIRO, M. M., Asst. Br. Mgr., Airtemp Div., Sidles Co., Lincoln, Nebr.	F. E. Prawl M. J. Stevenson	I. C. Baker L. L. Daubert
SPOERR, F. F., Air Cond. Engr., Carrier Dept., W. L. Thompson, Inc., Boston, Mass.	John James J. W. Millard	D. J. Edwards C. G. Norton (*Non-Member*)
TAPLEY, M. S., Htg. & Air Cond. Engr., Rudge & Guenzel Co., Lincoln, Nebr.	M. J. Stevenson C. V. Carlson	F. E. Prawl A. A. Luebs (*A. S. M. E.*)
TAYLOR, T. E., Air Cond. Engr., Control Equip. Co., Portland, Ore.	J. D. Kroeker E. C. Willey	H. K. Mead (*Non-Member*) F. M. DeTemple (*Non-Member*)
TAZE, E. H., Sales Engr., American Blower Corp., New York, N. Y.	F. R. Still R. G. Olson	Arthur Ritter W. E. Heibel
WAGNER, J. G., 1st Lt., Civilian Conservation Corp., Trenton, N. J.	D. C. Hawley (*Non-Member*) A. Johnson (*Non-Member*)	G. F. Bateman (*Non-Member*) T. A. Hendrick (*A.S.C.E.*)
WHITE, E. D., Engr., Minneapolis-Honeywell Reg. Co., Cleveland, Ohio.	R. A. Wilson F. A. Rodgers	L. T. Avery J. E. Wilhelm
ZUBER, OTTO, Chief Engr., Air Cond. Dept., Amana Society, Amana, Iowa.	A. G. Matousek L. J. Du Bois	H. A. Ware (*Non-Member*) J. M. Musser (*Non-Member*)

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ANGUS, F. M., Br. Mgr., General Refrigeration Sales Co., Cleveland, Ohio.

DRAKE, G. F., Development Engr., Barber-Colman Co., Rockford, Ill.

DUNNE, R. V., Engr., Carrier Corp., Newark, N. J.

GROOT, H. W., Chief Engr., The Home Comfortable, Inc., Louisville, Ky.

HALL, J. R., Chief Engr., U. S. Air Cond. Corp., Minneapolis, Minn. (*Advancement*)

HUMMEL, G. W., Field Engr. & Sales, Trane Co., La Crosse, Wis.

MCNEVIN, J. E., Mgr. & Vice Pres., G. P. Braid, Inc., Denver, Colo.

MOHN, H. L., Research Engr., Fitzgibbons Boiler Co., Inc., Oswego, N. Y.

MUNRO, G. A., Gen. Mgr., H. F. Munro & Sons, Philadelphia, Pa.

PETERSON, N. H., Mfgrs. Agent, Neil H. Peterson Co., San Francisco, Calif. (*Reconsideration*)

PHILIP, WILLIAM, Sales Engr., Dominion Radiator & Boiler Co., Toronto, Ont., Can. (*Reinstatement*)

ROSE, J. C., Air Cond. Engr., Buensod-Stacey, New York, N. Y.

SCHAFER, H. C., Htg. Dept., Iroquois Gas Corp., Buffalo, N. Y. (*Reinstatement*)

SIMONSON, G. M., Consulting Mech. Engr., San Francisco, Calif.

SOMMERS, W. J., Sales Repr., Sarco Co., Inc., Buffalo, N. Y.

VISSAC, G. A., Consulting Engr., Calgary, Alta., Can.

WOOD, C. F., Educational Dept., Delco-Frigidaire Cond. Div., Dayton, Ohio.

ASSOCIATES

ATHERTON, A. E., Director, A. E. Atherton & Sons, Pty., Ltd., Melbourne, Victoria, Australia.

DOWNES, A. H., Draftsman, Pernot & Rich, Los Angeles, Calif.

GULER, G. D., Sales Engr., Minneapolis-Honeywell Reg. Co., Philadelphia, Pa.

HALL, GEORGE, Gen. Mgr., Hyland, Hall & Co., Madison, Wis.

POLLOCK, C. A., Mgr., Dominion Electrohome Industries, Ltd., Kitchener, Ont., Can.

SEELBACH, HERMAN, JR., Sales, Minneapolis-Honeywell Reg. Co., Buffalo, N. Y.

THEOBALD, ART, Research Engr., Payne Furnace & Supply Co., Beverly Hills, Calif.

JUNIORS

AKERMAN, J. R., Engr., Phoenix Oil Co., Augusta, Ga.

ALBRECHT, H. P., Engr., Reinhard Bros., Minneapolis, Minn.

ARDEN, I. L., Engr. & Mgr., Gastonia Mill Supply Co., Charlotte, N. C.

BARRON, J. T., Asst. Engr., Hynes Elec. Htg. Co., Philadelphia, Pa.

BIRD, G. L. H., Chief Engr., Refrigeration (Birmingham) Ltd., Birmingham, England.

BRAYMAN, A. I., Htg. Contractor, Edw. Brayman, Boston, Mass.

CABOT, M. A., Design Engr., University of Kentucky, Lexington, Ky.

DIAMOND, D. D., Design Engr., Twin City Furnace Co., Minneapolis, Minn.

SPENCE, R. A., Steam Htg. Engr., Boston Edison Co., Boston, Mass.

STUDENTS

GREENBERG, IRVING, Design & Estimating, S. Greenberg, New York, N. Y.

SIDELL, P. A., Student, University of Illinois, Urbana, Ill.

NEWEST ADVANCE IN OIL HEATING

DELCO'S NEW *Rotopower Unit* *with the* Thin-Mix Fuel Control!

This latest development of Delco-Frigidaire cuts oil heating costs three ways...Saves fuel...Saves power... Saves upkeep

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The Delco Conditionair and Delco Automatic Furnace are also available for economical gas heat, using the faster, more radiant Delco Luminous Gas Flame.

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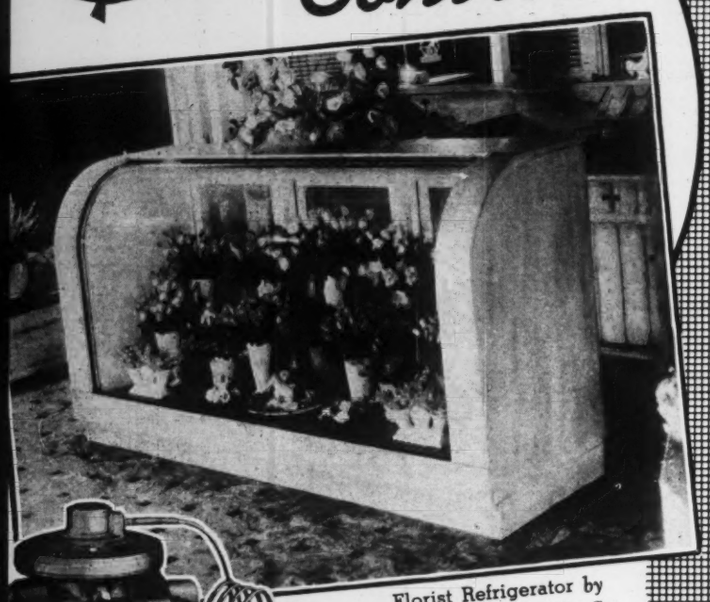

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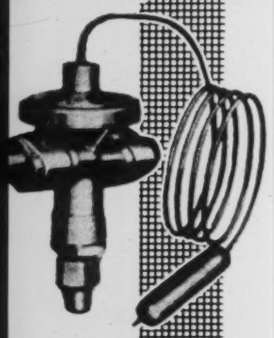

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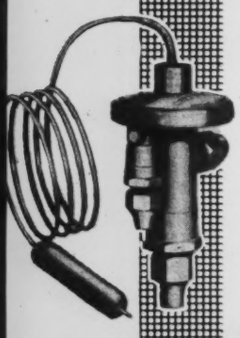
ansion Valve



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MILWAUKEE WISCONSIN

Schools and Study Courses

Gives Evening Instruction in Air Conditioning

Evening instruction in heating, ventilating and air conditioning will be offered by the Polytechnic Institute of Brooklyn, N. Y., during the regular 1937-38 school sessions, according to an announcement of Prof. E. F. Church, Jr., head of the department of mechanical engineering. Classes will be held on Wednesday and Friday evenings, with registration for the first session in heating and ventilating during the week of September 13, 1937. This course covers the basic elements of air conditioning with relation to the engineering design and selection of equipment for modern domestic and commercial systems. Problems will be given illustrating the application of various systems to residences apartments, office buildings, schools, factories and theaters.

Beginning in February, the course in air conditioning will deal with advanced considerations in the field of the previous course, and will present fundamental psychrometric principles and their application to cooling, humidifying and dehumidifying of air for central station or unit type systems. Sessions will cover theory of refrigeration, cooling load computations, air distribution and duct design, sound control, methods of surface cooling, bypass recirculation, and chemical dehumidification.

Both courses of instruction will be presented by John James, a member of the technical staff of the *American Society of Heating and Ventilating Engineers.*

To Offer Course in Air Conditioning

A practical training course dealing with the problems of selling, installing, and servicing air conditioning equipment will be sponsored by Fairbanks, Morse & Co. at the company's plant in Beloit, Wis. The school will be held during the entire month of October. Graduates of colleges, technical schools, and high schools who want to fit themselves for work in the air conditioning field are eligible to enter, it was announced. There will be no tuition charge.

Home Study Course on Oil Burners Announced

The University Extension Division of the Department of Education, Commonwealth of Massachusetts, 217 State House, Boston, has announced a home study course on practical oil burner engineering which has been prepared for service men, electricians, and engineers who install, service, plan or design.

Divided into two parts of eight lessons each, the subjects covered include oil fuels, combustion, atomization, oil burners, automatic control devices, oil fuel tests, heat measurements, calculation of heating systems and requirements, fans and blowers, drafts and chimneys, air conditioning tests, and estimating savings from heat insulation of walls and roofs.

Columbia Offers Evening Courses in Air Conditioning

The Department of Architecture of Columbia University, New York, N. Y., has announced that two evening courses in air conditioning will be offered during the winter session, which runs from September 23 to February 1. Instruction in one of the courses will be given by R. W. Waterfill of Buensod-Stacey Air Conditioning, Inc., and the other will be under the direction of C. A. Bulkeley, chief engineer of Niagara Blower Co.

Mr. Waterfill's course will include study of steam-air mixtures with relation to comfort cooling and commercial air condition-

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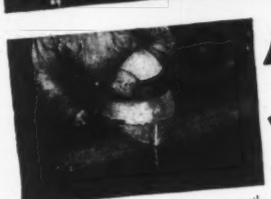
. . . Welded Outlets for Piping . . .



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3. On 2" outlets or larger the button is removed after installation. A full inspection of the inside of the joint is possible when WeldOlets or ThredOlets are used.

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400 Branch Outlets without Cutting or Threading the Main Pipe

STOP for just one minute and consider the savings made in installing 400 branch outlets on two 500' main steam lines, by using Bonney WeldOlets and ThredOlets. All cutting, threading and fitting of the main line was eliminated. No templets were used.

In addition to tremendous savings in installation, leak-proof joints of full pipe strength were obtained. Due to the funnel-shaped outlets of WeldOlets and ThredOlets, friction and turbulence were reduced to a minimum and free flow conditions were improved.

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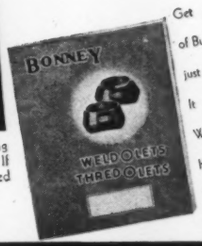

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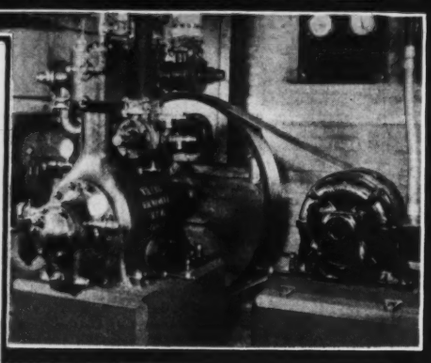

VILTER
Equipment
at
WORK

As a current example of "Vilter Equipment at Work" we point with pardonable pride to the recent Hollenden Hotel Air Conditioning System.

Called by engineers everywhere "The Sweetest Job of Air Conditioning in the Country" this modern installation is giving an extraordinary type of smooth-running efficiency, trouble-free service, and showing unusually low operating cost. It was planned and installed completely under Vilter supervision, and uses the latest types of Vilter Air Conditioning Equipment.

For Air Conditioning and Refrigeration of all kinds, you will profit by using Vilter Equipment.

VILTER
Manufacturing Company
2148 South First Street
Milwaukee - Wisconsin

ing, comfort cooling calculations, general study of equipment and design, and comparative analysis of refrigeration equipment and refrigerants used in connection with air conditioning.

Mr. Bulkeley's course will cover quantitative relations on performance of elements of air conditioning equipment and on installation assemblies in buildings; thermodynamics, heat transfer and fluid dynamics applied to heating, cooling, humidifying, dehumidifying and air movement, and to instruments or controls; and application to typical commercial equipment and building installations. The course is arranged to meet the needs of those concerned with sales, installation, or design of particular equipment or complete installations.

Book Reviews

Motor and Control Applications

"Motor and Control Applications," by George H. Hall, M.E. 1st ed. 1937. *259+ viii pp., 6x9 in., clothbound. Published by McGraw-Hill Book Co., 330 W. 42nd St., New York, N. Y. Price,* $3.00.

This book is intended to assist the machine designer in the selection of motors and controls, forming a link between the motor and control as produced by the motor manufacturer, and the tool or machine designed by the machine builder. It is the author's aim not to deal with the details of motor design, but to present as concisely as possible the characteristics obtainable from the various types of motors and the performances that may be secured from them by the selection of suitable control.

The work is quite complete in its scope and in its discussion of the subjects with which it deals. Of particular interest are chapters on fan and blower applications (10 pp.), pump and compressor applications (12 pp.), and motors for quiet operation (7 pp.).

In addition to the chapters just mentioned, the following are included: Determination of machine requirements, direct current motors, direct current control, squirrel cage induction motors, wound rotor induction motors, synchronous motors, single phase motors, multispeed motors, protective control devices, control accessories, stopping devices, motor enclosures (general), motor and controller enclosures (for hazardous conditions), high frequency induction motors, shaftless motors, integral speed reducers, connecting the motor to its work, machine tool applications, woodworking applications, hoist and chain applications, miscellaneous applications and *N.E.M.A.* manufacturing standards.

Oil Fuels and Burners

"Oil Fuels and Burners (With Special Reference to Automatic Domestic Types)," by James A. Moyer, State Director of University Extension in Massachusetts. 1st ed. 1937. *375+ ix pp., 6x9 in., clothbound. Published by McGraw-Hill Book Co., 330 W. 42nd St., New York, N. Y. Price,* $4.00.

The author has endeavored to meet the need for an informational manual on oil burners (particularly of the automatic type for the heating of residences and apartment houses) for the use of service men, salesmen, mechanics, electricians, and engineers who are engaged in servicing, planning, and designing in this field. The related subject of heat insulation of buildings is also given attention, and a chapter comprising a brief discussion of air conditioning is included.

The content comprises chapters on oil fuels, combustion, atomization, burners, automatic control devices, oil fuel tests, heat measurements, calculation of heating systems, calculation of heating requirements, estimating savings from insulation of walls and roofs, fans and blowers, draft and chimneys, and air conditioning tests.

It's the "V" ANGLES that make

ARCO AIR FILTERS

. . . *Last Longer*
. . . *Hold More Dirt*

1. The "V" angle construction makes the air change directions quickly and scrub against the sticky sides of the filter. 2. As the dust impinges on the sides of the filter it absorbs the sticky coating and in turn becomes a dust catcher. 3. The space between the two wafers acts as an added storage space for the dust that is collected. Remember there is one of these efficient Arco Air Filters for every air conditioning or ventilating job. They are light, odorless, inexpensive, and won't drip oil even at 180 degrees F. The absolutely uniform construction leaves no weak spots for the dirty air to get through. Write today for complete details.

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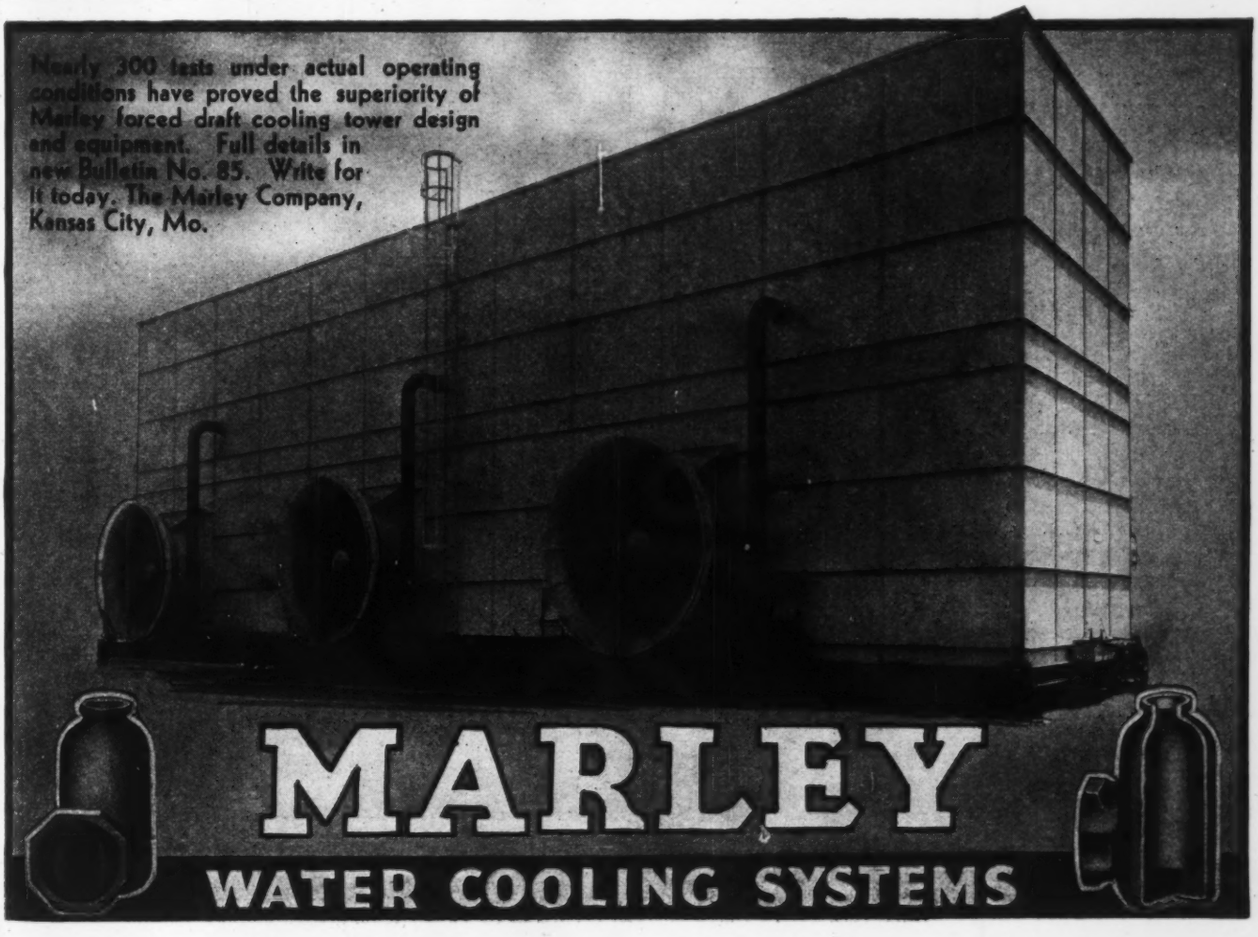
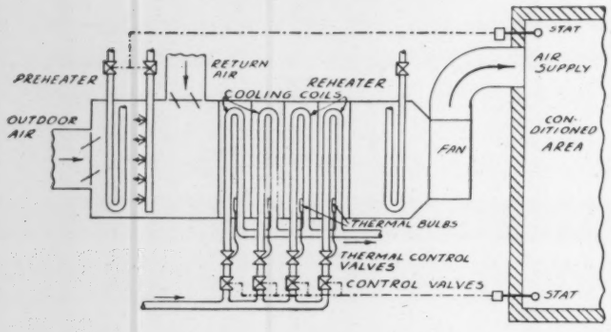
40 West 40th Street, New York, N. Y.

ARCO AIR FILTERS


Nearly 300 tests under actual operating conditions have proved the superiority of Marley forced draft cooling tower design and equipment. Full details in new Bulletin No. 85. Write for it today. The Marley Company, Kansas City, Mo.
MARLEY
WATER COOLING SYSTEMS

UP TO DATE PRACTICE IN AIR CONDITIONING



THIS Auditorium Conditioning Corporation System (*) readily meets the requirements of a wide range of air conditioning installations for human comfort.

Due to accurate control of dehumidification responsive to varying conditions in the room the intensity of air treatment at all times varies with the load and although all air is passed through the dehumidifier the temperature of the room is always within the comfort range.

The system is comparatively cheap to install and operating costs are reasonable.

Representatives of our licensees in all sections of the country will be glad to discuss the advantages of this system.

In addition to the above, Auditorium Conditioning Corporation has many Systems (*) capable of solving your air conditioning problems. These are described in literature available from our Licensees. Why not get acquainted with these systems?

LICENSEES

American Blower Corp.
Detroit, Mich.

Buffalo Forge Co.
Buffalo, N. Y.

Carrier Corp.
Newark, N. J.

Frick Co.
Waynesboro, Pa.

General Electric Co.
Schenectady, N. Y.

J. O. Ross Engineering Corp.
New York, N. Y.

The Cooling & Air Conditioning Corp.
Division of B. F. Sturtevant Co.
Hyde Park, Boston, Mass.

Westinghouse Electric & Manufacturing Company
East Pittsburgh, Pa.

York Ice Machinery Corp.
York, Pa.

*) *Auditorium Air Conditioning Systems are covered by many issued patents and pending applications*

AUDITORIUM CONDITIONING CORPORATION

New York Office—17 East 42nd Street . New York

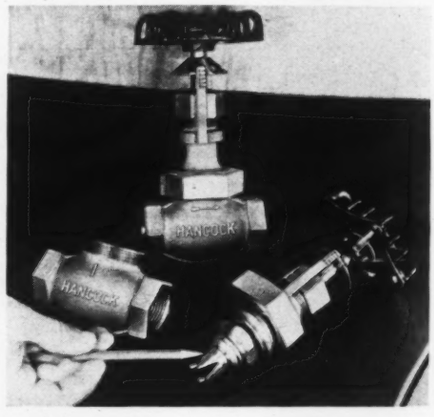
Equipment Developments

For your convenience in obtaining more information about any of this equipment, see coupon on page 160. Add the new products and companies listed here to your Directory Section which you received in your January, 1937, HEATING, PIPING AND AIR CONDITIONING and thus keep your records of sources of supply up to date throughout the year . . . Single asterisk (*) indicates equipment not listed in Directory Section; double asterisk (**) equipment and manufacturer not listed.

Flow Control Valves Redesigned

No. 1286—Redesigned bronze "Flocontrol" valves are in reality a combination flow instrument and shutoff valve.

The new valves retain the basic proportional flow or straight-line flow characteristic of the original valves, according to the maker. They incorporate "superhard" stainless steel valve seats and discs of 500 Brinell hardness, and it is claimed that these hard valve seats and discs will smash nails, pipe turnings, welding chips and boiler scale without leaving a trace on either seat or disc.



They are intended for use in industrial and utility plants for maintaining closer manual control on oil burner lines, on water treating systems, on cooling water lines to turbines and air compressors, and for continuous blow-down.—Hancock Valve Div., Manning, Maxwell & Moore, Inc., Bridgeport, Conn.

Develops Line of Condensate Return Units

No. 1287—New condensate return outfits for handling radiations up to 40,000 sq ft, boiler pressures up to 120 lb, are especially adapted for automatically returning condensate from steam radiators, steam jacketed kettles, pipe coils and other steam heated apparatus, to low and high pressure boilers, says the maker. Condensate is collected in receiver, from which pump forces it back to boiler. Care has been exercised to utilize properly sized receivers so that water level in boiler is not seriously affected and condensate, without being retained too long, is pumped back as near boiling point as possible.

Pumps are either reciprocating or centrifugal, depending upon the rating of the outfit, and are complete with motor with full overload and low voltage protection. Receiver tanks are of rust resisting, copper bearing steel to insure long life. A two-pole float switch with copper float furnished with and mounted on the receiver automatically starts and stops pump according to

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FRICK
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And proves it with this array of fine buildings, recently Frick-equipped.
These jobs include several "firsts" in Tulsa, also the largest air conditioning contract ever let in the Southwest, requiring over 1000 tons of refrigeration.
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DEPENDABLE REFRIGERATION SINCE 1882
FRICK CO.
WAYNESBORO, PENNA. U.S.A.
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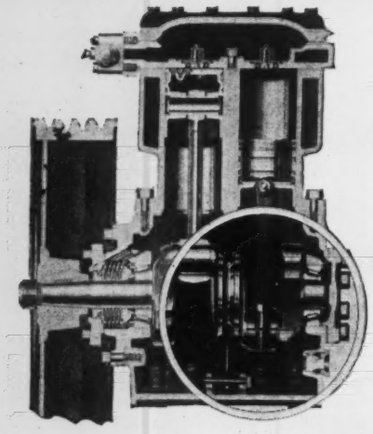
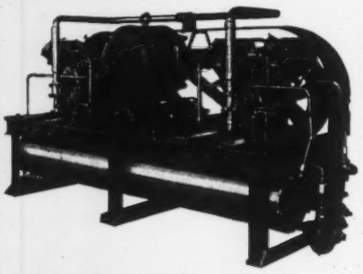
REX-WELD "Super-Service"
VIBRATION ABSORBERS
for Air-Conditioning Machinery
● Here's the ideal solution to the compressor noise problem! Installed in the suction and discharge lines, these Rex-Weld Flexible Metal Tubing Units effectively dampen the vibration and put an end to the annoying hammering—surely, safely and economically . . . Produced by an exclusive patented process which forms the flexible tubing corrugations without weakening the special, non-porous, non-crystallizing bronze alloy. Supplied complete with capped ends ready for sweating onto standard water tubing. Write for Bulletin RW-4 and Data Sheet. Send us your specifications for prices.
CHICAGO METAL HOSE CORPORATION
(Formerly Chicago Tubing and Braiding Company—Est. 1902)
Maywood, Illinois (Chicago Suburb)

CURTIS
Condensing Units



"CENTRO-RING" LUBRICATION—
—only
CURTIS COMPRESSORS
have it



THERE'S just one moving part in Curtis "Centro-Ring" lubrication—a simple, free running ring that picks up oil from the crankcase and distributes it by centrifugal force under positive, uniform pressure to all points of lubrication—no gears—no plungers—nothing to get out of order or require service.

Engineering superiorities of this kind have built the Curtis reputation for reliability, long operating life and low cost of upkeep.

The Curtis line contains 41 air-cooled units, from 1/6 HP to 5 HP, inc., and 45 water-cooled units from 1/3 HP to 30 HP — a size and type to balance most any installation you might make. Curtis has built condensing units for 15 years and has made engineering equipment for 82 years — a record of success that in itself is sufficient warranty of the high-quality of Curtis units of today.

CURTIS Engineering Refinements

- Centro-Ring — one piece, positive pressure lubrication
- Timken Tapered Roller Bearings
- V-type radial compressor
- Water-jacketed compressor heads and cylinders
- Built-in oil separator with automatic return
- Drop forged, heat treated crankshafts and rods
- Balanced syphon, bellows seal
- Automatic water valves

Represented in Canada by
Canadian Curtis Refrigeration Co., Ltd.
20 George St., Hamilton, Ont., Can.

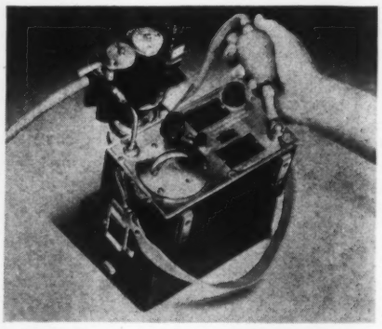
CURTIS REFRIGERATING MACHINE CO.
"Builders of Condensing Units since 1922"
1930 Kienlen Ave., St. Louis, U. S. A.

water level in the receiver.

Provision for automatic feed water make-up with boiler level pump control can also be furnished with these outfits when specified. This arrangement is for use in laundry, dry cleaning or similar installations, where some of the steam is exhausted into the atmosphere or used for other purposes, and does not condense and return to the boiler. In such cases, the outfit will automatically furnish fresh make-up water as well as return to the boiler such condensate as may be available. In such installations, the pump is automatically controlled by the level of the water in the boiler, which is maintained at the proper point continuously, and new water is added only as needed.—Goulds Pumps, Inc., Fall St., Seneca Falls, N. Y.

Indicators Detect Dangerous Gas Mixtures

No. 1288—New gas indicators, developed for quickly and easily detecting explosive or irrespirable gas-air mixtures, are known as the "UCC" gas indicators. These instruments are designed to fulfill the requirements of public utilities, gas plants, shipping companies, petroleum companies, paint and lacquer companies, solvent manufacturers, fire departments and insurance companies for locating underground leaks, establishing purge-end points, and determining hazardous conditions in manholes, oil tankers, oil storage tanks, holds of ships, sewers, conduits and tunnels.



The combustible gas indicator model is for the detection of combustible gases or vapors. A graduated meter scale indicates by direct reading whether combustible gases are present, and, if they are, whether concentrations are above, within, or below explosive limits. The all-service model combines three units in one. It shows the presence of combustible gases, indicates an oxygen deficiency, and, in addition, includes a toxic chamber for determining the presence of carbon monoxide and hydrogen sulphide in dangerously toxic concentrations.—The Linde Air Products Co., 30 E. 42nd St., New York, N. Y.

Announces Burst-Proof Steam Hose

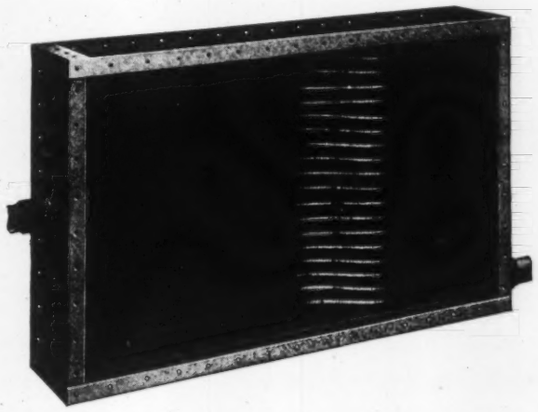
No. 1289—When the new burst-proof steam hose illustrated fails it will leak but nothing can cause it to burst except an external blow severe enough to cut the wire reinforcement, according to the manufacturer. It is recommended for 200 lb working pressure in all sizes, with a safety factor of 15 to 1 in larger diameters and up to 35 to 1 in small diameters.

Tube (A) and insulating layers (C, E) are made of a special heat-resisting rubber compound. The reinforcement of high tensile steel wire (B, D) cannot burn out and deteriorate. Outside of the wire braid and underneath the cover is embedded an additional braid of asbestos cord (F), which protects the cover from burning. Completing the construction is a heavy black rubber cover which is compounded with an antioxidant to give it age-resisting qualities.

Available in sizes ranging from ⅜ to 2½ in. — B. F. Goodrich Co., 548 S. Main St., Akron, Ohio.

AEROFIN

Heat-Exchange Surface



A SIZE and TYPE for every REQUIREMENT

Aerofin's complete line of fan system heating and cooling surface offers the best possible choice in the selection of equipment adapted to meet any requirement. Its quality of workmanship, thorough reliability and efficient performance have been proved in thousands of installations.

Aerofin is noted for its exclusive features such as the Crown Orifice to stop scale clogging, water units with removable headers to permit easy cleaning of tube interiors and direct expansion cooling units with Centrifugal Header for refrigerant distribution.

The constant study of the industry by Aerofin engineers assures the user of the latest equipment as required by modern practice in air conditioning.

Try Aerofin on your next job. Ask the home office or any of our branch offices for complete data. They will also be glad to assist you in solving special problems.

AEROFIN is sold only by Manufacturers of Nationally Advertised Fan System Apparatus. — List upon Request

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The Master Insulation for Heating Systems...

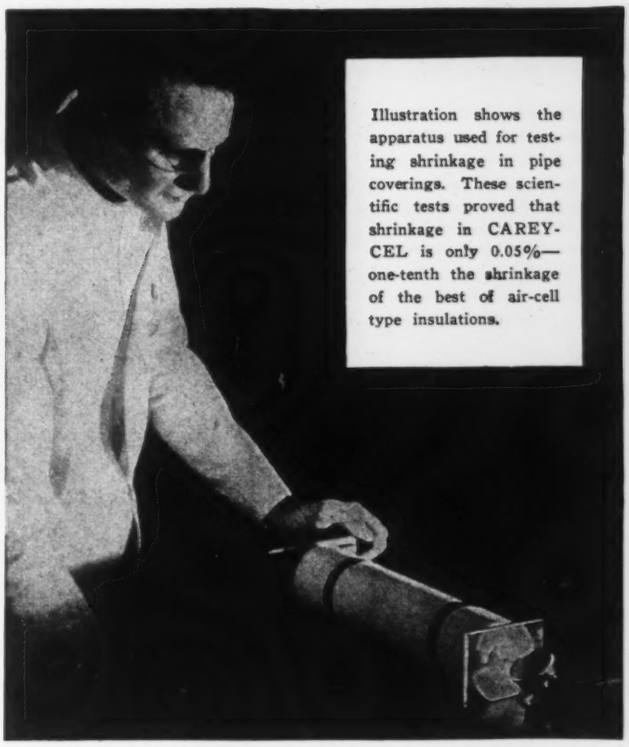

Illustration shows the apparatus used for testing shrinkage in pipe coverings. These scientific tests proved that shrinkage in CAREYCEL is only 0.05%—one-tenth the shrinkage of the best of air-cell type insulations.

... Careycel ASBESTOS INSULATION

The cross-sectional illustration below shows the smallness of air spaces in Careycel as compared with ordinary air-cell. This is the principal reason for its higher efficiency and is made possible by the exclusive Carey process of "cellizing" the asbestos sheets.

Consider these advantages of CAREYCEL over air-cell insulations when figuring future jobs:

CUTS HEAT LOSS 30%

Impartial tests at Mellon Institute conclusively demonstrated that heat loss through CAREYCEL is 30% less than through an equal thickness of ordinary air-cell.

MINIMUM SHRINKAGE

Shrinkage is negligible. Eliminates unsightly, heat-wasting cracks at joints.

BETTER APPEARANCE

CAREYCEL provides a smooth, even, symmetrical surface—gives a more finished, workmanlike appearance.

Although CAREYCEL has these distinctive, outstanding advantages, it SELLS IN THE SAME PRICE RANGE as air cell.

Satisfy customers—build business—increase profits with CAREYCEL. Samples and positive proof of CAREYCEL superiorities sent on request. See nearest Carey Distributor, or write. Address Dept. 25.

THE PHILIP CAREY COMPANY
Dependable Products Since 1873
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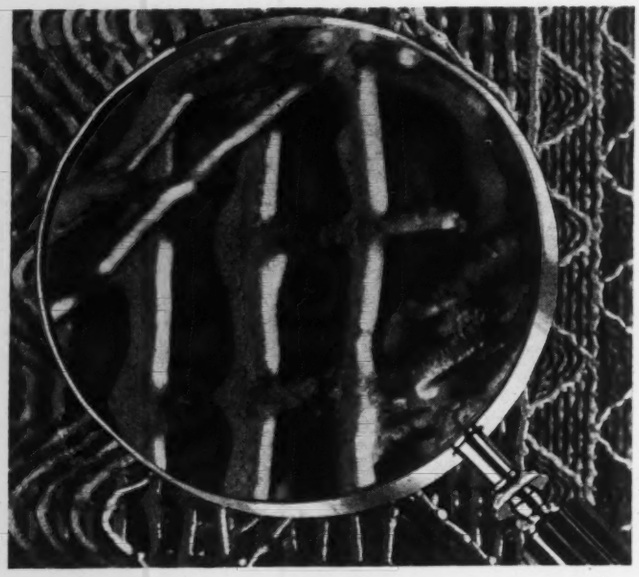
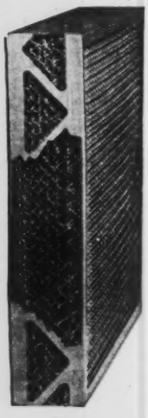


WHY AIR-MAZE
Assures Clean Air . . .

The microphotograph shows a cross-section of an AIR-MAZE air filter panel in operation. Observe that dust, dirt, grit and pollens **pyramid evenly,**—that the unique element "captures" all foreign particles, yet no clogged areas can form.

Such scientific design and precision construction assure exact and uniform progressions of density in the element. This means that there can be no weak or restrictive spots to raise velocity and thereby lower efficiency.



AIR-MAZE ADVANTAGES:

- 99.83% air cleaning efficiency.
- Easily cleaned and charged.
- No replacements needed.
- Very low pressure drop.
- Long, economical service.

For top notch performance PLUS low cost maintenance, specify AIR-MAZE Cleanable Air Filter Panels. Write for Bulletin BPAN-86.
See AIR-MAZE in the A. S. H. & V. E. Guide Book

AIR-MAZE CORPORATION
820 Huron Road, Cleveland, Ohio

AIR-MAZE
CLEANABLE
AIR FILTER PANELS

Develops New Alloy Steel Welding Rod

No. 1290—New gas welding rod, named the "Airco No. 1 High Ductility Alloy Steel," has been designed for increased ductility and general improvement in quality of both single and multi-layer steel welds. The manufacturer states that a feature is its ability to withstand considerable heating without burning.

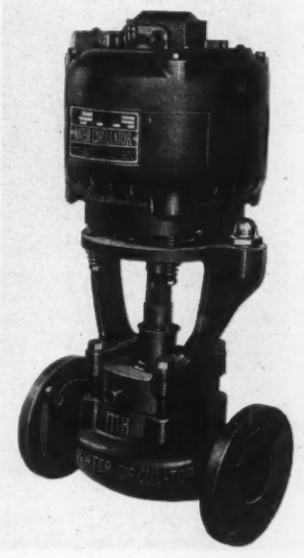
Physical characteristics reported by the maker are:

1. Free bend ductilities of single layer welds, range from 20 to 30 per cent depending upon the composition of the steel.
2. Free bend ductilities of multi-layer welds, up to 40 per cent on low and medium carbon steels.
3. Ultimate tensile strengths in excess of 60,000 lb per sq in.
4. Specific gravity of welds, 7.80-7.86.
5. Charpy impact values on keyhole notched specimens at 70 F, from 15 to 30 ft-lb.
6. Rockwell hardness of weld metal from B60 to B85, depending on carbon content of base metal and type of weld.

—Air Reduction Sales Co., 60 E. 42nd St., New York, N. Y.

Water Circulators Have Several Features

No. 1291—New water circulator, packless sealed and precision built, has been developed to speed up circulation on hot water heating systems and may be used alone for this purpose, or in conjunction with the manufacturer's flow valve to provide fully automatic summer-winter hot water control. Advantages are quiet operation, leak-proof due to the packless seal which does not permit the oil to enter water or vice versa. Standard with built-in thermal overload cutout, it requires low starting current and offers no radio interference.



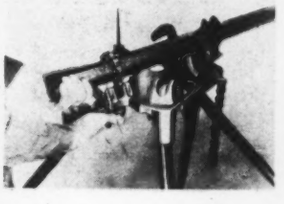
The circulator is scientifically lubricated to insure trouble-free performance; the oil supply can readily be renewed.

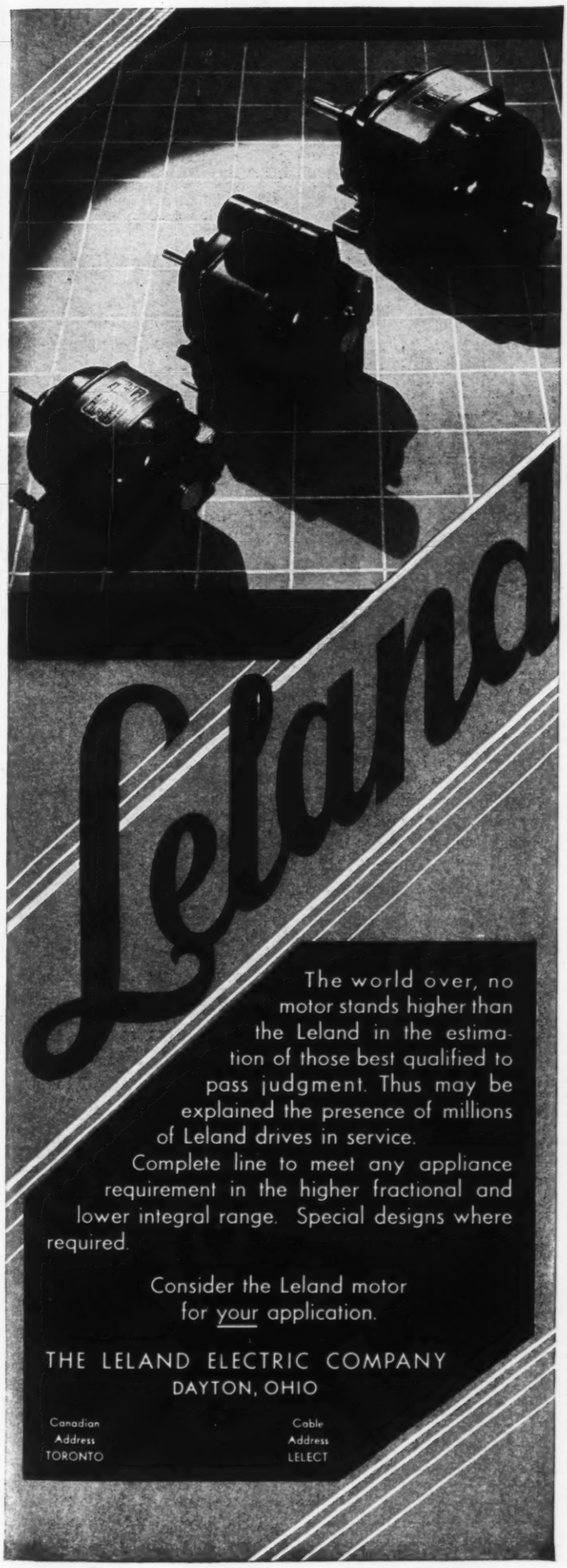
It is offered in sizes to fit 1, 1½, 2, and 3 in. pipe mains. Although 110 volt, 60 cycle motors are standard, the circulator is available for 220 volts, 60 cycles and 110 or 220 volts, 25, 30, 40 or 50 cycles, or for 115 or 230 volts d-c.

—Minneapolis-Honeywell Regulator Co., 2701 Fourth Ave., S., Minneapolis, Minn.

New Pipe Vise and Bender, Chain Pipe Tong, and Box Wrenches

No. 1292—New "Vulcan" vise stand is a portable unit combining complete stand, chain pipe vise and pipe bender. The base is made of malleable iron, well-reinforced. Legs are 1 in. pipe with upset feet punched for fastening to the floor, if desired. Legs

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Complete line to meet any appliance requirement in the higher fractional and lower integral range. Special designs where required.
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SO THAT'S WHERE OUR HEAT GOES— WELL — YOUR IDEA'S TOO EXPENSIVE, I'LL PUT WING FLOODLIGHT HEATERS UP THERE TO SEND THE HEAT DOWN HERE —
PROJECTS THE HEAT TO THE WORKING AREA
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DEPENDABLE

BLAST COILS

A COMPLETE LINE
FOR
ALL APPLICATIONS

* * *

High and Low Pressure
STEAM HEATING
WATER COOLING
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DIRECT EXPANSION

Approved by leading architects, engineers and contractors.

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McQUAY, Inc.
MINNEAPOLIS MINN

fold for carrying and are held together by tie chain. Vise jaw is of tool steel, hardened and tempered. Stand is finished in orange enamel with black legs, weighs 35 lb, and takes pipe ⅛ to 2 in. Pipe bender handles pipe up to ¾ in.

The same manufacturer has also announced a new chain pipe tong, drop forged from alloy and high tensile steel and heat treated, known as "Vulcan Supertong," with 50 per cent greater strength. Finished in gray enamel. Jaws have blue panel with bright faced edges. Eight sizes for pipe ⅛ to 18 in.

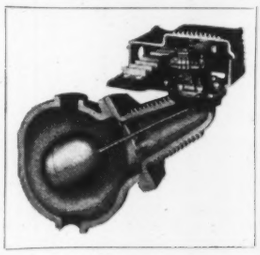
Also recently announced are new heavy pattern hex-box wrenches designed for severe service. Their comparatively thin head walls are particularly efficient in close quarters, says the maker. Openings range from 1¼ to 3⅛ in. Dropforged in two lines—carbon steel wrenches and chrome-molybdenum "Superrenches."

Carbon wrenches are finished in black enamel (baked-on) with bright heads. "Superrenches" are chrome plated with the faces of heads buffed bright.—J. H. Williams & Co., 75 Spring St., New York, N. Y.

Boiler Control Unit Does Two Jobs

No. 1293—A new boiler control unit which is a combination pump control and low water cut-off in one unit is designed for use with steam pressures up to 150 lb.

These two controls have distinctly separate operating levels, and the differential between the two circuits is adjustable, as is the differential between the "off" and "on" operation of the controls.

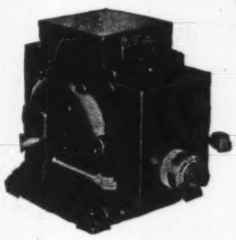


A flexible metallic bellows transmits float power to the outside switch mechanism, and eliminates the use of stuffing boxes. The pump control circuit is designed to turn on and off an electrically driven boiler feed pump in accordance with the requirements of the boiler. At a lower level, a second separate circuit will shut off the oil burner or stoker in case of a fuse failure in the pump circuit.—McDonnell & Miller, 400 N. Michigan Ave., Chicago, Ill.

Facilitates Installation of Room Coolers

No. 1294—The "Mobilcart" has been devised to facilitate the

Holtzer Cabot



CUSTOM BUILT MOTORS

Are Designed and Engineered
To Fit the Job
Assuring Long, Quiet,
Dependable Service

●

Let Us Help
You Solve Your
Motor Problem

THE HOLTZER-CABOT
ELECTRIC COMPANY
BOSTON, MASS.
MOTOR-SPECIALISTS

For Best Air Conditioning
RESULTS

INSULATE COOL AIR DUCTS
AND AIR WASHING ROOMS
WITH
NOVOID CORKBOARD

YOU CAN deliver cool air more efficiently—more economically—when you insulate air ducts with Novoid Featherweight Corkboard. Made of pure cork, Novoid forms a lasting barrier to the passage of heat and effectively checks condensation.

Novoid is available in four convenient sizes: 12x36, 18x36, 24x36, and 36x36 inches. Thickness ranges from one to six inches. It is light in weight, structurally strong, and may be applied easily and quickly to round, square, or rectangular ducts.

For insulating air-washing rooms, Novoid Standard Corkboard is equally efficient and prevents the costly infiltration of warm air. For more complete details, mail the coupon below or write Cork Import Corporation, 330 West 42nd Street, New York City.

BRANCH OFFICES: Chicago, 400 West Madison Street; Boston, 248 Boylston Street; Philadelphia, 1524 Chestnut Street.

NOVOID CORKBOARD INSULATION

MAIL THE HANDY COUPON TODAY

Cork Import Corporation
330 West 42nd Street, New York City

Please send me full information on Novoid Corkboard Insulation and its application to air ducts.

Name .....

Street .....

City and State .....

Bellevue-Stratford Hotel, Philadelphia

PROVIDE BALANCED HEAT IN BELLEVUE-STRATFORD

World-Famous Philadelphia Hotel Carries Out Webster Heating Modernization Program

PERFECT COMFORT FOR GUESTS

Philadelphia, Pa.—The world-famous Bellevue-Stratford Hotel took advantage of the opportunity to provide better heating service for guests by installing the Webster Moderator System of Steam Heating early in 1936.

Because the Webster Moderator System supplies continuous mild heat to each of the Bellevue-Stratford's 2,518 radiators, there is no on-and-off heating with its alternate "cold 70" and overheating. Room temperatures are adequate at all times regardless of changes in outdoor weather conditions.

F. J. Malloy, Chief Engineer of the Bellevue-Stratford, states: "Our chief concern has always been to provide perfect heating comfort for guests regardless of the severity of the weather. With the Webster Moderator System, we have achieved that result."

Although better accommodations for guests was their principal objective, performance records show that the Bellevue-Stratford substantially reduced the cost of heating by installing the Webster Moderator System.

Prior to modernization, Webster engineers estimated that coal consumption would be reduced 150 tons during March and April, 1936, the period set aside for the working out of operating procedure. Just how accurate this estimate proved to be is shown by the fact that actual fuel savings during March and April, 1936, amounted to 154 tons.

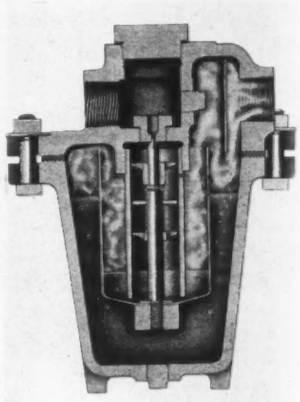
These before-and-after facts point the way to maximum comfort and economy in heating new buildings as well as in modernization of existing installations. Consult your architect, engineer or heating contractor. Or address WARREN WEBSTER & CO., Camden, N. J. Pioneers of the Vacuum System of Steam Heating Representatives in 60 principal U. S. Cities—Est. 1888

WEBSTER MODERATOR CONTROL "Control by the weather"

handling of the "Mobilaire" portable unit air conditioner. A combination elevator and truck, it makes easy the loading, unloading, and transporting of the units from place to place, room to room. No lifting, no shoving, no straining, no damage to expensive floors, or floor coverings, no danger of chipping walls or damaging window sills, are among its advantages, says the maker.—Westinghouse Electric & Mfg. Co., Mansfield, Ohio.

Offers New Bucket Steam Trap

No. 1295—New straight line, bucket type steam trap makes it unnecessary to disturb piping or connections when inspection or cleaning is desirable, the manufacturer points out.



The trap body is of molychrome iron, bucket of drawn copper, valve and seat of hardened stainless steel, discharge tubes and guides of bronze. Trap ratings are scientifically determined and traps guaranteed to discharge condensate at rated capacities, the maker advises. — Cochrane Corp., 3161 N. 17th St., Philadelphia, Pa.

Designs Water Valve for Air Conditioners

No. 1296—Electric valve designed especially for the control of water supply to air conditioners and humidifiers has a novel scheme for attaching BX cable: The lower half of the coil housing is empty, and allows the wires from the BX cable to be pushed into this space after connections are made with the coil leads. Then the BX sheath is held in place by the housing connector.

Another feature, says the maker, is the absence of the usual click of the plunger when the valve opens or closes. The plunger has a "composition tip" to close the valve quietly. The upper end of the plunger travel is silenced by a rubber pad. The valve seat and plunger are removable for cleaning.

Connections may be for ⅛, ¼, or ⅜ in. pipe, with orifice 1/16 or ⅛ in. diameter.

Valves will control also gas, oil, steam, and other liquids.—Supreme Electric Products Corp., 99 Mt. Hope Ave., Rochester, N. Y.

Low Water Cut-Off Improved

No. 1297—Features of improved low water cut-off include quick installation, flexible mounting, alarm or signal circuit, easy accessibility, and improved mercury switch. Can be installed according to the *A. S. M. E.* boiler code or in the gauge glass fittings of the boiler with the resulting saving in labor and material.

Designed to fit standard half-inch gauge glass openings, an installation feature is that the lower fitting may be readily removed if there is not enough room to swing the entire assembly. A copper tube between mountings can be cut to accommo-

Why

We Don't Urge Using A Steel Oil Boiler

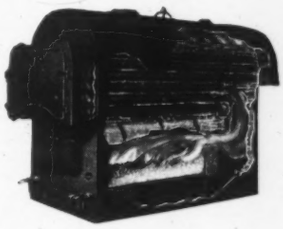
(Yes We Make Them)

Don't you feel, that the boiler to use, is the boiler that's best for that use? Steel, if steel is the one. Cast iron, if such is so.

Now suppose we made only steel boilers. Then we would just plain have to do a strong-arm selling job on steel for every place and purpose.

The Cast Iron made in twin sections so it can be passed through any average door. "No tearing out" necessary. Used extensively for replacements.

Making both cast iron and steel boilers as we do, our only concern is, that you get the one *best for the job.*



The Welded Steel Burnham ranges in capacities from 1800 to 42500 sq. ft.

So that's why we don't urge your using either steel or cast iron. But we do urge fitting the job with the boiler that best fits it. Yours is the advantage of our being in an unprejudiced position.

Burnham Boiler Corporation

Manufacturers of Heating Equipment Since 1873

IRVINGTON, NEW YORK, ZANESVILLE, OHIO

Export Department

116 Broad Street, New York City

Burnham Boiler

Heat

JUST WHERE YOU WANT IT

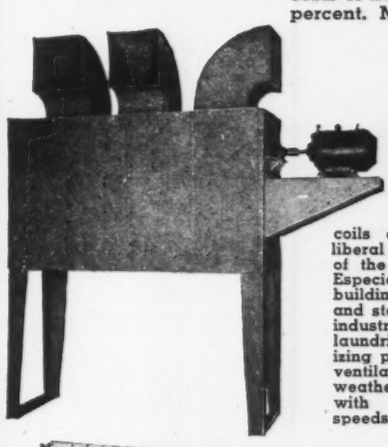
with NATIONAL FAN AND BLOWER CORP. CHICAGO, ILL.

UNIT HEATERS

The name "NATIONAL" on a heater is assurance of careful engineering, high efficiency, low operating cost and long dependable service. A full range of sizes in each of three types, for all requirements.

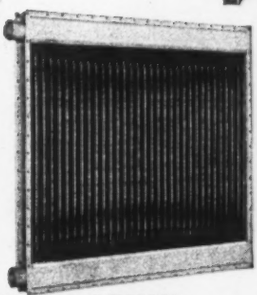
SUSPENDED TYPE UNIT HEATER

Designed for overhead installation. They furnish the necessary volume of heat where wanted, raise the temperature a degree a minute and cut costs of heating as much as 30 percent. Made in 8 sizes, with wide variety of motor speed.



BLOWER TYPE UNIT HEATER

For distributing heat uniformly over large areas. Air is heated by passing over the coils and is discharged, in liberal volume, over the heads of the workers in the room. Especially effective in high buildings. Adaptable for fog and steam elimination in such industries as packing plants, laundries, dairies and galvanizing plants. Also effective as ventilating units in warm weather. Made in four sizes, with a variety of motor speeds.



BLAST HEATER

Of exceptionally high efficiency. Tested at 1,000 lbs. hydrostatic pressure. Guaranteed to operate with steam pressure as high as 150 lb. gauge. Available with one or two rows of tubes in direction of air flow. Same coil as used in National Heaters. Made in all standard sizes, to suit your requirements.

100% AGENCY FRANCHISE

We have an exclusive agency franchise on the complete National line of air conditioning equipment, including blowers, propeller fans, air washers, etc. State territory desired.

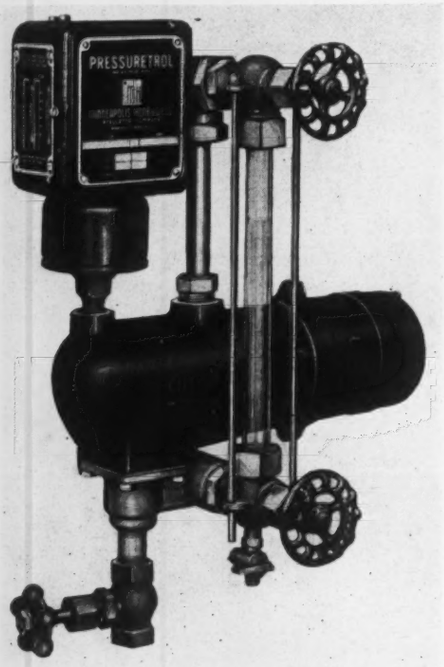
SEND FOR THIS CATALOG

NATIONAL FAN AND BLOWER CORPORATION

540 West Washington Boulevard Chicago, Ill.

date various center dimensions and shaped to allow for any slight misalignment. The switching mechanism may be removed without dismounting the instrument from the boiler.

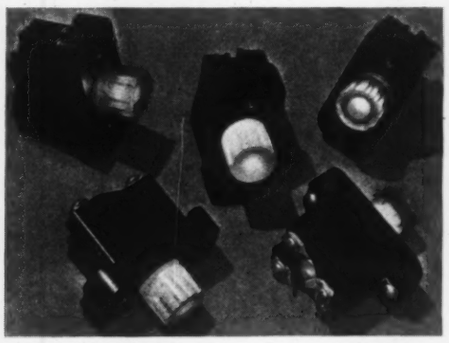
Cut-off is designed for use with pressures not exceeding 10



lb per sq in. The electrical rating is 10 amp at 110 volts and 5 amp at 220 volts. A manual reset is obtainable at slight additional cost, necessitating manual attention to restore burner operation.—Minneapolis-Honeywell Regulator Co., 2701 Fourth Ave., S., Minneapolis, Minn.

Combines Push Button, Indicating Lamp

No. 1298—A small, standard duty push-button station with an indicating lamp in the center of the translucent button has been developed. Through the combination, it needs only about one-half the space required by the two devices when they are mounted

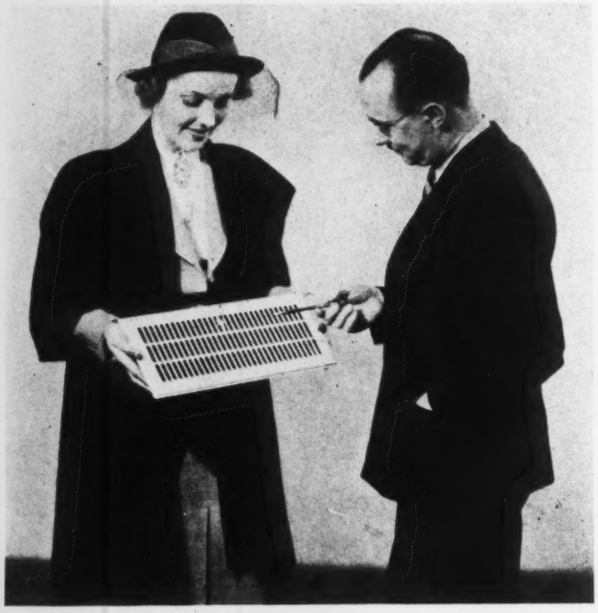


separately. It is available with buttons in white, green, blue, red, clear, and amber.

The indicating lamp is operated through a small control transformer which, because of its diminutive size, can be mounted readily in or on a machine.—General Electric Co., 1 River Rd., Schenectady, N. Y.

Announces New Electrodes

No. 1299—New coated electrodes especially adapted for alternating current welding, known as "Crucible-Weld," are available in sizes from $\frac{3}{32}$ to $\frac{5}{16}$ in. diameter, and are made in two grades. In addition a special $\frac{1}{16}$ in. electrode is available for use with very low current values. All offer the advantages of arc sta-



She Wants to Know—

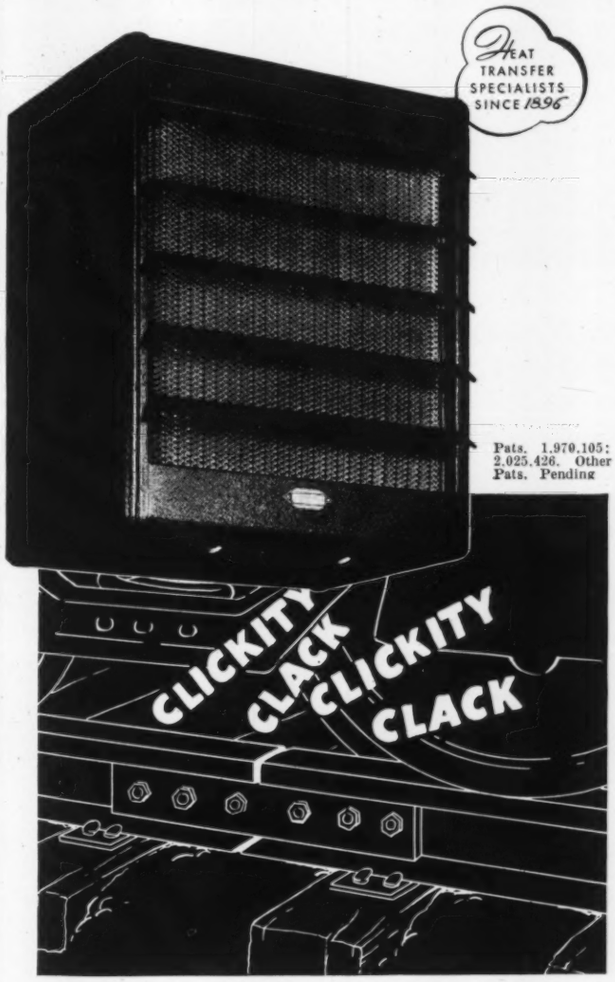
The average woman whose husband has money enough to build or buy a good house today is a keen-minded shopper. She wants to *know* what kind of registers she's getting—from the standpoint of both appearance and dependability. She's particular about the heating or air conditioning system.

Show her the AUER "Classic"—or the other popular AUER faces. They are styled for fine modern interiors and built for lasting satisfaction. AUER makes a complete line of registers, intakes, and grilles for both warm air and air conditioning. Ask for latest catalog.

THE AUER REGISTER COMPANY, 3608 PAYNE AVENUE, CLEVELAND, OHIO

AUER DISTINCTIVE REGISTERS & GRILLES For Air Conditioning *and* Gravity

RAILROAD TRACKS
are Protected with
EXPANSION JOINTS



Pats. 1,970,105; 2,025,426. Other Pats. Pending

FEDDERS Unit Heaters
Are Protected Against Self-Destruction by

. . . Exclusive Full-Floating mountings which permit free movement of the heating element vertically in cabinet, and also protect it from torsional stresses. Tubes having 3 to 1 streamline ratio and individual convoluted fins allow each tube to "give" laterally, thus relieving differential expansion stresses within the element. This also allows the use of curved, non-diaphragmatic headers, assuring a stable heating element. Fedders convoluted fins are saddled over sides of tubes, to eliminate heat expansion stresses between fin and tube, thus maintaining a permanent thermal bond. Accordion design of fins eliminates stresses between bonding points.

Quieter operation and handsome, sturdy mono-piece cabinets make Fedders Unit Heaters ideal for commercial as well as industrial applications. Write for Bulletin 572.

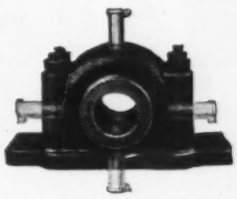
FEDDERS MANUFACTURING CO., BUFFALO, N. Y.

All Season Air Conditioning Units—Blast Heating and Cooling Surfaces

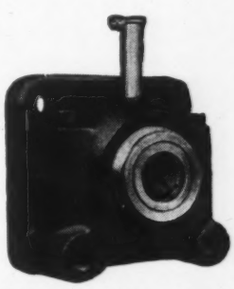
Air Conditioning Blowers demand these *Quiet* bearings by *Randall*

Standard—For ordinary bearing service where a single reservoir supplies sufficient lubrication.

D.R.O.R.—For heavy duty or high speed applications. Has two reservoirs and greater oil capacity.



D.R.O.R. Universal—Ball can be turned so that mounting can be made in any of four positions.



D.R.O.R Flange—Designed especially for mounting on cabinet type air conditioners and unit heaters.

No air conditioning equipment can possibly operate satisfactorily with noisy bearings, because these sounds are picked up, amplified and carried throughout the duct system. Randall bearings have completely solved this problem because they run quietly and without metallic bearing noise.

They are self-aligning, self-lubricating, and operate with minimum running torque. Operating without attention for long periods, they are dependable and have extremely long life.

With the exception of the "Standard" with single reservoir, Randall Pillow Blocks are equipped with the D.R.O.R. (double reservoir oil return) feature, which provides large oil supply in the two reservoirs.

Randall Pillow Blocks are now standard equipment on a majority of all air conditioning and blower units. Write for full information.

Pacific Coast Stocks

Utility Fan & Mfg. Co. Los Angeles, Cal.

C. W. Marwedel, San Franscisco, Cal.

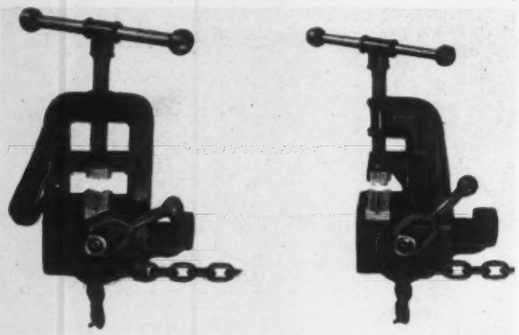
Randall GRAPHITE PRODUCTS CORP.
Dept. 910 609 W. Lake St., Chicago, Ill.

bility, low spatter loss, high tensile strength and high ductility of deposit, the maker states.

One grade is especially suitable for down hand butt welding because of the high melting rate and ability to operate at high current values with low spatter loss. The slagging characteristics permit their use on sloping surfaces where the slope may be as much as 30 deg. The other grade is recommended for general purpose work, horizontal, vertical and overhead welding, and is adapted for fillets, single or multiple pass.—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Post Vises Grip Tight

No. 1300—Vises which can be tightly locked to any column, pillar or post up to 10 in. in diameter are now available for use



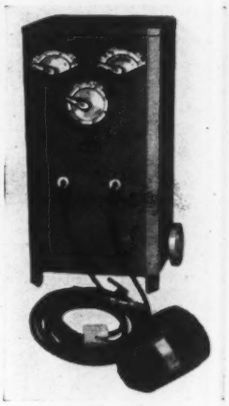
where a piping job is to be done and it is more convenient to bring the vise to the job than to cut the pipe at the shop.

A snap of the chain, a turn of the locking handle and the vise is set on post or column as securely as though it were riveted in place, says the maker.

The new series consists of two types, the side open post vise which holds pipe from ⅛ to 2 in., and the hinged post vise which holds pipe from ⅛ to 2½ in. Both models are equipped with ¾ in. pipe bender.—Armstrong Mfg. Co., 303 Knowlton St., Bridgeport, Conn.

Welder Features Many Current Settings

No. 1301—An a-c welder with separate voltage and amperage controls, enabling the operator to select the most desirable voltage for the amperage used on any job, is known as "Dual Control," furnishes —says the maker—practically unlimited current settings.

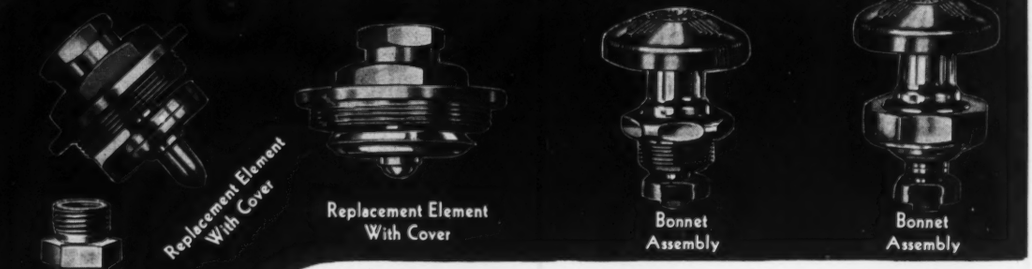


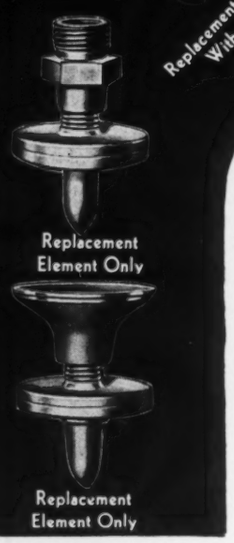
Built in three sizes with a current range from 10 amperes to maximum output that makes possible welding of sheet metal or heavy metal, the welder is equipped with wheels and a handle so that it is portable. All three sizes have the same cabinet dimensions and vary in weight from 235 to 350 lb. Furnished standard for 220 volts.—Miller Electric Mfg. Co., 905 N. Meade St., Appleton, Wis.

Radiator Air Valve Completely Adjustable

No. 1302—Completely adjustable steam venting control from maximum open to minimum closed is provided by the visual adjustment on the new line of "Warco" air and vacuum valves, it was announced last month.

MODERNIZE WITH MILVACO





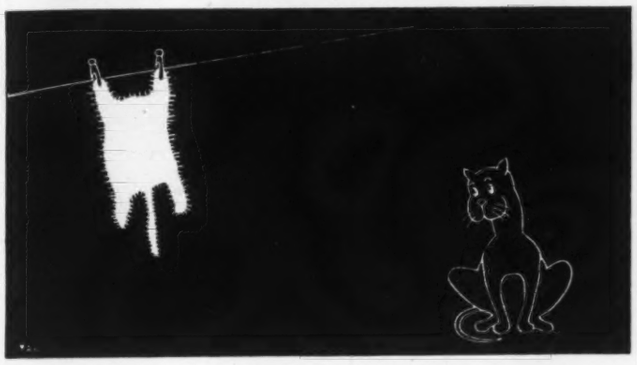
SPECIALIZED
MODERNIZATION UNITS

MILVACO Specialized Modernization Units are easy to install and easier to adjust. No special tools or equipment necessary. They restore new unit operating efficiency to the trap or valve in which they are installed at only a fraction of the cost of a complete new unit.

There is a MILVACO Modernization Unit to fit every radiator valve and trap in popular use today. Consult your nearest MILVACO representative or write for particulars.

MILWAUKEE VALVE COMPANY
MILWAUKEE Manufacturers of WISCONSIN
MILVACO Heating Specialties and APPROVED Bronze Valves

THERE'S MORE THAN ONE WAY TO SKIN A CAT!

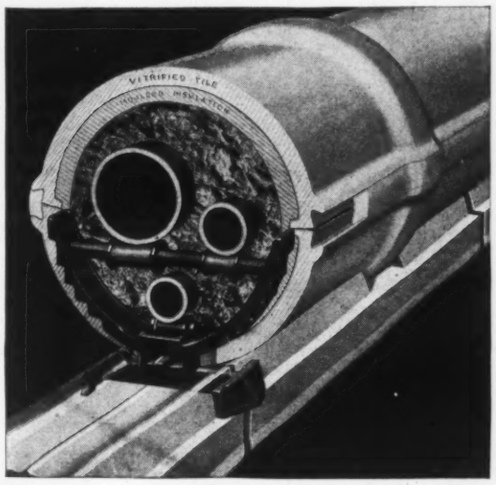


YOU can skin a cat and make a fur coat—but it won't be much good. You can skin an underground steam conduit job—and save money to pay out in steam losses. Ric-wiL never did it that way. In material and manufacture, the Ric-wiL idea is that permanent efficiency is worth what it costs. If you want a makeshift system, you don't want Ric-wiL. If you want the lowest priced system that will actually operate ECONOMICALLY—you do want Ric-wiL. We back that up with THOUSANDS of jobs all over the country.

Water repellance, heat retention, underdrainage, alignment, ease of assembly—all these vital requirements are completely met. Ric-wiL keeps steam lines over 90% efficient. Various designs in Tile or Cast Iron to meet all conditions—famous Dry-paC Waterproof Asbestos Insulation. Submit your underground steam problems to Ric-wiL. Ask for complete catalog with Dry-paC sample for testing.

The Ric-wiL Co., Union Trust Bldg., Cleveland, Ohio
New York San Francisco Chicago
Agents in principal cities

Standard Ric-wiL Conduit is assembled on the job—materials and workmanship can be easily inspected.



REGISTERED IN U. S. PATENT OFFICE

RIC-WIL

CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

NOW ALL ALLOY

It Practically
STOPS WRENCH REPAIRS

That RIDGID housing guarantee saves you at least 75% of your wrench repairs and having wrenches out of commission—a valuable economy.

But the RIDGID gives you also tremendous strength because it's all-alloy now with alloy handle and chrome molybdenum jaws, heel jaw replaceable, hook jaw full-floating and made with handy pipe scale. Adjusting nut spins easily in every size, 6″ to 60″. Altogether, a wrench with satisfactions you can't find in any other.

Ask Your Jobber

THE RIDGE TOOL CO., ELYRIA, O.

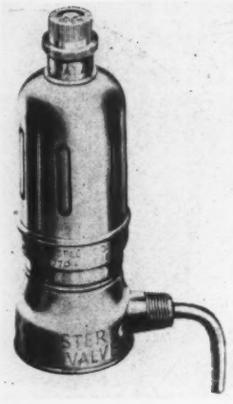
RIDGID PIPE TOOLS

You ought to have this RIDGID Compound Leverage Wrench in every kit

Makes one man as strong as 6—pays for itself in salvaged fittings on the first tear-down job.

The adjustable cap on top of the valve is set to vent the correct amount of steam. A positive locknut, directly below the adjustable cap, allows the correct setting to be fastened in place and prevents tampering or accidental change.

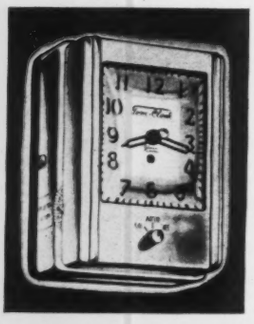
With this feature, the amount of heat in any individual radiator can be accurately regulated regardless of distance from the boiler, the maker says. It also offers the advantages of completely "stepless" control.



The adjustable cap is supplied on a complete line of air and vacuum valves for one pipe steam systems. —W. A. Russell & Co., Rm. 1734, Grand Central Terminal Bdg., New York, N. Y.

Announces Two Clock Controls

No. 1303—Two new "Tem-Clocks" for automatic electric clock control of lowered night temperature for use in conjunction with "Temtrol" (heat anticipating room thermostat) have just been announced.



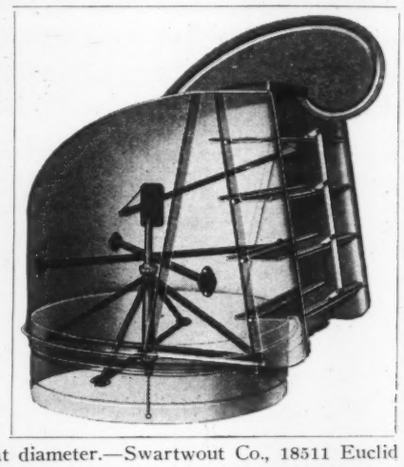
Type 772 is for service on all types of heating plants controlled by Temtrol. It provides three control positions which may be set easily by means of a selector knob: (1) Automatic control, whereby the electric clock operates a switching mechanism to lower automatically temperature at night and restore it to day level in the morning. (2) Continuous high or day level temperature. (3) Continuous low or night level temperature.

Type 774 is designed especially for stoker installations. It performs the day-night control functions of the standard type and in addition, operates a timed interval, hold-fire switching mechanism which may be set for one to 15 minute firing periods every 30, 60 or 90 minutes as selected by the installation man. It eliminates the need for a synchronous electric motor to operate a hold fire switching mechanism.

Both types are operated by silent, synchronous, self-lubricating "Telechron" motors.—Penn Electric Switch Co., Goshen, Ind.

Rotary Roof Ventilator Improved

No. 1304—New rotary roof ventilator for industrial and commercial building application offers improved features, according to the maker. Foremost of these is the new stainless steel, fully enclosed, dust-tight, oil-less ball bearings on which the head turns with the wind. Interior construction is of streamlined steel tubing welded into a one piece head frame of minimum friction resistance. The body is designed along modern lines resulting in a low pleasing appearance with a slightly increased area in the discharge opening.



It is made in sizes from 12 to 72 in. throat diameter.—Swartwout Co., 18511 Euclid Ave., Cleveland, Ohio.

Refrigeration Costs are

LOWERED....

Mundet Cork Insulation for Air Conditioning

. . . when "Jointite"* Cork Insulation is used for low temperature requirements. Due to our method of selecting, grading and fabricating the pure Cork, you get Cork insulation at its maximum efficiency when you specify "Jointite."

"Jointite" insulating products include Corkboard (in the correct thicknesses for air conditioning service) and Moulded Cork Pipe Covering in standard lengths and thicknesses.

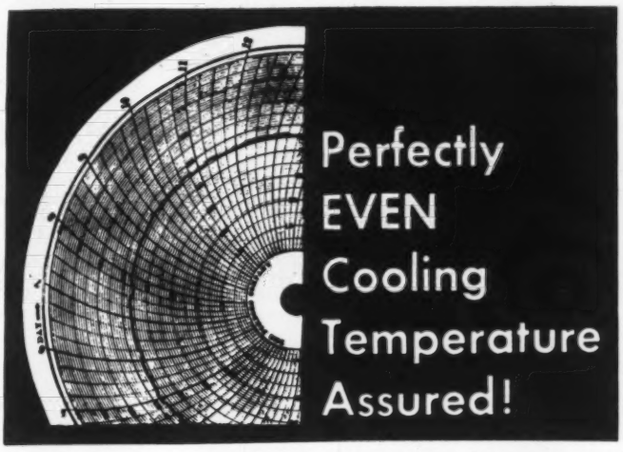
Our engineering department will be glad to cooperate with you in the preparation of specifications and estimates. Catalog sent on request.

Machinery vibration is permanently isolated with Mundet Natural Cork Mats. Ask for details.

*Trade Mark Reg. U. S. Pat. Off.

MUNDET CORK CORP.

450 SEVENTH AVE., NEW YORK, N. Y.
Offices in Principal Cities



With a Patterson Freon Water Cooler in your air conditioning system, there will always be enough superheat in the suction line to insure positive operation of the thermo expansion valve. Consequently, there will be no fluctuations to cause uneven water temperatures and varying conditions in the conditioned space.

To provide a permanent segregation of refrigerant charges for each compressor, Patterson Coolers are designed for as many circuits as there are compressors.

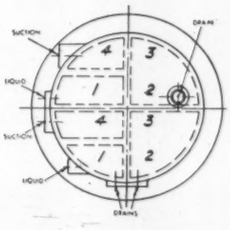
The U-tube construction permits each tube to expand and contract independently of the others. Therefore no joints will be opened, permitting leakage of Freon. And as the tubes are made of copper, they cannot pit or rust.

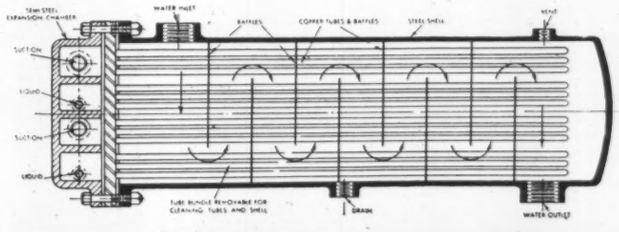
Another point—Patterson Coolers return the oil to the crank case of compressor without the necessity of any special oil draining connections.

These are but a few of the important features of Patterson Freon Water Coolers explained in our booklet. Write for a copy.

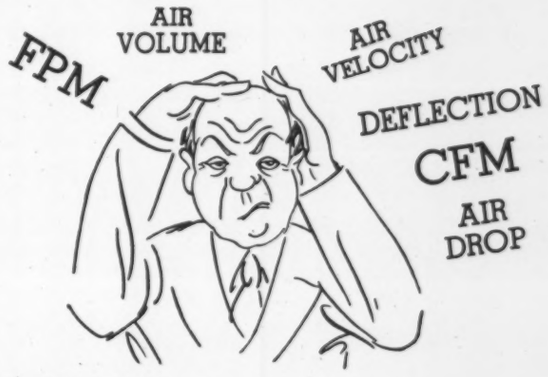
The Patterson-Kelley Co. Inc.

107 WARREN STREET
EAST STROUDSBURG, PA.
New York Office: 101 Park Avenue





Patterson Freon Water Coolers

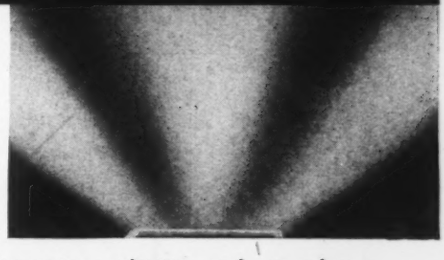


SURE THERE ARE HEADACHES — BUT TUTTLE & BAILEY ENGINEERS HAVE HELPED A LOT OF 'EM

In a laboratory which, as nearly as possible, recreates actual air conditioning installation conditions, Tuttle & Bailey Engineers have succeeded in taking a lot of the headaches out of air conditioning work.

The question of proper distribution of air from the outlet is studied carefully with smoke: facts and not theories alone are available.

The Airline Grille is a development of laboratory study. Close mesh, attractive and strong, it is a scientifically designed outlet, of the fixed air deflection type, which will accomplish certain types of air distribution better than any other grille we know of.



Here is an actual "smoke test" experiment in the T & B laboratory. This is an Airline Grille with 3-way deflection. Such tests as this are used to determine air throw, air drop and deflection at varying velocities and air volumes.

Send for Latest Catalog

T & B
TRADE MARK

TUTTLE & BAILEY
Incorporated
NEW BRITAIN, CONN.

NEW YORK CHICAGO BOSTON PHILADELPHIA

Cooperative GLF Office Building Ithaca, N. Y.

NEW GLF BUILDING GETS MODERN HEATING

American Warming & Ventilating Co. Installs Webster Moderator System

Concealed Webster Radiation

Ithaca, N. Y.—Built during 1936, the modern, five-story office building of the Cooperative Grange League Federation stands as a tribute to the careful planning and sound business management of the Cooperative officials.

Working on a limited budget, officers of the G L F sought long life and low maintenance cost in selecting equipment.

Study of the performance records of Webster Systems of Steam Heating in scores of installations convinced Grange officials that the Webster Moderator System was the soundest possible investment for the heating of their new building.

The Webster Moderator System provides **balanced** heating service. Heat is delivered continuously to all radiators and the entire building is comfortably heated regardless of the severity of the weather and without overheating in mild weather. Heating costs are held continuously at a minimum.

The installation includes 70 concealed Webster System Radiators. The grilles of these "out-of-the-way" concealed radiators harmonize with the modern interior and increase available floor space. All cast-iron radiators used in the building are equipped with Webster Three-Point Valves to insure maximum flexibility.

Two Webster Boiler Protectors guard against accidental low water in the low-pressure heating boilers. If water is lost from a boiler accidentally, the Webster Boiler Protector automatically maintains emergency level and provides the operator with definite indication that water has been lost.

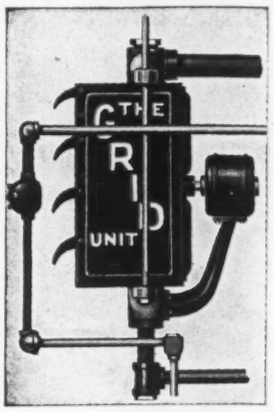
The Webster System was installed in the G L F Building by the American Warming & Ventilating Co., well-known heating contractors of Elmira, N. Y. The building was designed by Arthur N. Gibb, a leading Ithaca architect.

These before-and-after facts point the way to maximum comfort and economy in heating new buildings as well as in modernization of existing installations. Consult your architect, engineer or heating contractor. Or address WARREN WEBSTER & CO., Camden, N. J. Pioneers of the Vacuum System of Steam Heating Representatives in 60 principal U. S. Cities—Est. 1888

WEBSTER MODERATOR CONTROL "Control by the weather"

Humidifier Developed for Unit Heaters

No. 1305*—Recently developed "Grid" humidifier may be installed in front or directly behind the unit heater, permits operation of both elements as a unit or operation of the humidifier individually. The humidifier operates at temperatures higher than the room atmosphere, which causes quick absorption of the humidifying fluid by the room air, according to the manufacturer.

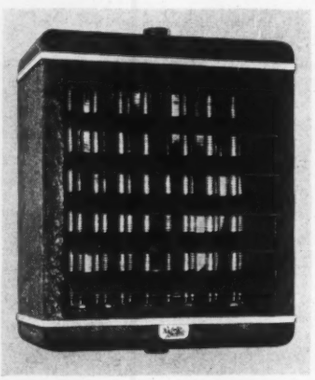


No air lines or extensive piping are necessary, and waste from the humidifier is handled directly to the sewer. Control is by humidistat.

Twelve models are available, the smallest having capacities from 0.04 to 0.16 lb of water per hr with steam pressure ranging from 1 to 10 lb; the largest has capacities from 2.8 to 10.8 lb of water per hr over the same steam pressure range.—Unit Heater & Cooler Co., 1002-1024 Third St., Wausau, Wis.

Unit Heaters Are of De Luxe Design

No. 1306—New "de luxe" design of propeller fan unit heaters have casings styled to harmonize with modern store and shop interiors, yet ruggedness of construction makes the units equally suitable for garage and factory use. Ease and flexibility of application are accomplished by providing six sizes of units, the maker points out, each with a choice of three heating elements of varying capacities.



For heating large interiors, this company's line includes a complete range of sizes in centrifugal fan "Giant Heaters"; these are available in floor, wall mounted, and ceiling suspended types, and may be had either with or without the "Thermadjust" temperature control damper. The manufacturer's principle in unit heaters is to handle large volumes of air with relatively small temperature rises, and to discharge the air at high velocity to the desired area. For this method, mixability of the air, the elimination of stratification, and economy of fuel are claimed.

Made by John J. Nesbitt, Inc., these units are distributed by Warren Webster & Co., 17th and Federal Sts., Camden, N. J.

Announces Development in One Pipe Steam Venting

No. 1307—Large port, low pressure venting for automatically fired one pipe heating systems is offered by the new "Ideal Fast-Venting" system, which includes the "Arco-Detroit Multiport" for radiators (at right) and the "Hurivent" for mains (at left). The makers state that the automatic modulator, patented key to the new design, has eliminated the obstacles to large port venting without sacrificing any advantages.

Short cycle heating control complicates venting, it is stated, because it further reduces the on period of the burner. With a relatively short run of main riser to vent, the radiator in that room is likely to heat up rapidly and cause the thermostat to reach its set point; thus the burner may be cut off before more remote radiators have a chance to vent and heat up. It is to

WHERE WRENCH QUALITY DETERMINES THE LEADERSHIP THERE IS AN OVERWHELMING PREFERENCE FOR TRIMO

made of Chrome Molybdenum Steel . . . drop forged . . . easily replaceable parts.

Greater safety and efficiency make Trimo a preferred wrench throughout industrial plants everywhere.

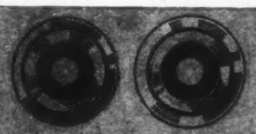
New TRIMO ALLOY PIPE WRENCH
HEAT TREATED
STEEL ALLOYS

Look for the Red Tag always attached to every Trimo . . . identifies Trimo quality.

TRIMO SAWING VISE SET

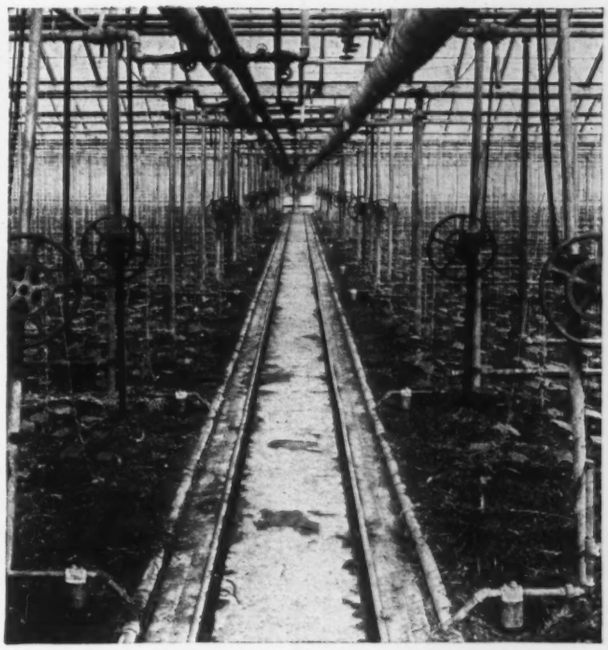
Increases efficiency and lowers costs on every thin wall copper tubing job. Every cut 100% square. No wasted tubing to be thrown away.

TRIMONT MFG. CO., INC., ROXBURY (BOSTON), MASS.



TRICROME CUTTER WHEELS
Thin Blade Special Analysis Steel — alvared to resist rust. Identified by Red Hub. Fit Trimo Barnes Saunders Cutters.

TRIMO PIPE CUTTERS
Trimo, Barnes and Saunders type.



A MATTER OF LIFE or DEATH

In a greenhouse, plant life is dependent on the heating system. Even one failure may mean the death of every plant and a total financial loss. For this reason greenhouse owners carefully scrutinize the performance record of steam traps when they buy.

The popularity of Anderson Super-Silvertop steam traps in the greenhouse industry is due to their dependability and the reserve capacity which is so frequently necessary to meet sudden and severe weather changes.

Although your problem may not be the heating of greenhouses, you do want dependable traps with reserve capacity. You can solve your heating and processing steam trap problems with Anderson Super-Silvertops—there is a size and style for every heating requirement. The Anderson reputation and over half a century of steam trap manufacturing experience are your guarantee.

Get in touch with our nearest distributor or write direct for further information about Anderson Super-Silvertop, the only complete steam trap.

THE V. D. ANDERSON CO.
1949 W. 96th STREET • CLEVELAND, OHIO

Anderson **SUPER-SILVERTOP**

EXPERT HANDS

Day after day we make pumps—nothing but pumps. Our designers, engineers, craftsmen, salesmen and service men specialize on pumps — exclusively. We know pumps—how to design and make them, and what they will do. Years of devotion to doing one thing well have given us expert hands, directed by expert minds. Every Pomona Pump is the product of specialists. The known dependability of Pomona Pumps created a world-wide demand, to meet which we operate the largest and best equipped factories in the turbine pump industry—with expert hands—devoted exclusively to making deep well turbine pumps and Niagara low lift drainage pumps.

POMONA PUMP CO.

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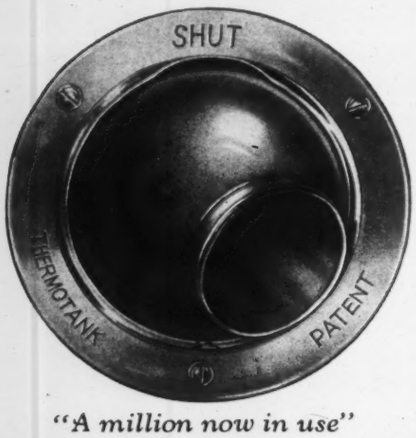
POMONA TURBINE PUMPS



"A million now in use"

MORE POSITIVE DISTRIBUTION and GREATER DIFFUSION of AIR With the Thermotank

PUNKAH LOUVRES

Because the Thermotank Punkah Louvre adjusts instantly to any desired position, air convection, direction and distribution are positive. Stratification of air cannot exist when the Thermotank Punkah Louvre is used. This directional air conveyor insures complete diffusion of air, with no dead "spots" within the large area of its distribution. The Thermotank Punkah Louvre prevents drafty currents; CONTROLS the direction and convection of air. It requires little wall or ceiling space, and each louvre can be directly controlled, thus insuring the desired amount of air distribution within its range. You get the air where you want it, ONLY with the Thermotank Punkah Louvre.

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KELVIN-WHITE COMPANY

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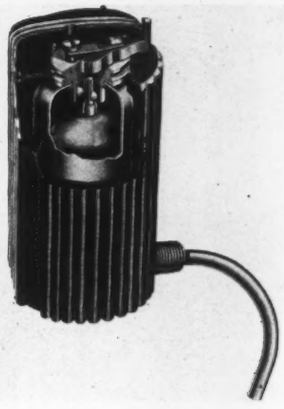


correct this and other difficulties that the new system has been developed.

The "Multiport" carries a manual adjustment which reduces its maximum port opening by any amount needed to compensate for the size and location of the radiators. Thus venting throughout the entire system may be balanced. The range of adjustment is large because the initial port opening is large, and the adjustment is said to be proportional to the movement of the adjusting arm.

Other advantages claimed for the new construction are that the muffling chamber will stop objectionable hiss and sizzling. Because the large port is fully open only below pressures of approximately 2 oz, it avoids the danger of flooding. If loose scale should prevent valve closure under water, the moulded jacket prevents spitting. This jacket is securely fastened to the valve, protects its working parts, and prevents tampering with the operating mechanism.

The styling of the "Multiport" is new and modern. The lustrous jet black jacket is trimmed with chromium in the form of four vertical stripes.—Detroit Lubricator Co., 5900 Trumbull Ave., Detroit, Mich.

Booklets, Reports and Papers

Welded Piping Design

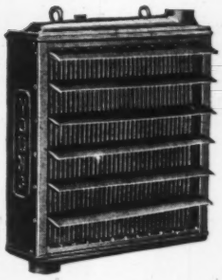
Complete handbook information on design and layout of piping for welded connections is contained in "Design of Welded Piping," a 200 p., 6x9 in. booklet containing over 100 figures and tables, published by The Linde Air Products Co., 205 E. 42nd St., New York, N. Y.

Subjects treated of interest to engineers are fundamentals of welded joint design; welding metallurgy; standard welded pipe connections; design data on welding cast iron, galvanized iron, stainless steel and non-ferrous piping; advantageous layout; fabrication and erection considerations; welded anchors and supports; and welding speeds.

Features of interest to architects and draftsmen include standard joint designs; typical headers, expansion bends and riser connections; anchors and supports; a typical heating system layout showing symbols for welds; two sample time-saving specifications; and principal dimensions of pipe, flanges, and welding fittings.

Briefly Noted

Air Conditioning Standards—Revised 1937 edition of the "Chicago Standards for Comfort Air Conditioning," the last revision of which was dated October, 1936, incorporating a few changes in requirements, some rewritten paragraphs, and some new ones. Chairman of the committee that prepared the standards is John Howatt, 228 N. LaSalle St., Chicago, Ill.



UNIT HEATERS – BLAST HEATERS – COMMERCIAL HEAT TRANSFER SURFACES

MODERNIZE YOUR HEATING PLANT WITH Young UNIT HEATERS

Either the suspension or floor models will provide a most satisfactory and economical method of heating all types of industrial buildings. Young Blast Units and Commercial Units are designed to operate in conjunction with central fan systems for either heating, cooling, or air conditioning. All Young heating and cooling units are available in many sizes and capacities to fill the most exacting specifications.

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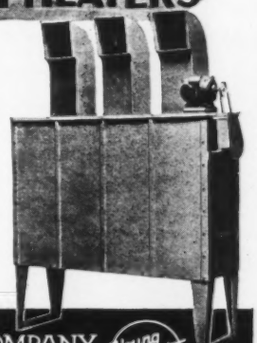

YOUNG RADIATOR COMPANY
RACINE, WISCONSIN
Young Quality

HYDRON METALLIC BELLOWS
are used as
CONTROL ELEMENTS

in temperature and pressure control devices, and for liquid or gas seals of compressors and pumps. Due to the hydraulic method of manufacture, Hydron metallic bellows are long-lived. Internal friction is eliminated, because Hydron bellows are formed from a one-piece tube by a series of cold-drawing operations and the final hydraulic forming of the bellows. We produce Hydron metallic bellows to individual specifications.

Write for our new catalogue containing important engineering data on all types of bellows.

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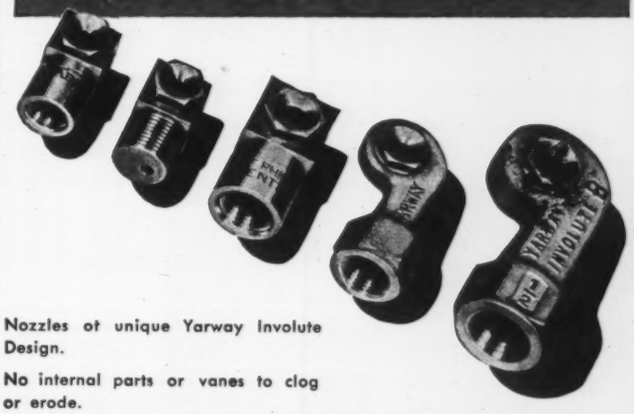
564 East First Street Boston, Mass.

PROTECTION

Your best protection against inferior merchandise and unreliable service is the reputable plumbing and heating supplier near you. It is to your advantage to deal with him exclusively.

Kuhns Brothers Co.
Malleable Iron Fittings Co.
K MIF

AIR CONDITIONING NOZZLES



Nozzles of unique Yarway Involute Design.

No internal parts or vanes to clog or erode.

Insure trouble-free air washing service.

Sizes and types for all requirements.

Many large users—installations total more than 5 million gallons per minute in water cooling and air conditioning service. Write for Bulletin N-615.

YARNALL-WARING COMPANY
MERMAID AND ANDERSON STS.
PHILADELPHIA

YARWAY

ONLY 4° WARMER AT ROOF THAN AT BREATHING LEVEL

With the SENSATIONAL AIRBLANKET UNIT HEATER

BLANKETING STREAM

The AIRBLANKET Definitely Reduces Heating Cost

. . . because only the AIRBLANKET brings below breathing level comfort with above breathing level economy.

This amazing unit heater does not rely upon fins, louvers, deflectors or other ordinary devices to retain the heat in the working zone. An entirely different principle, using a high velocity of cool air to roll the stream of warm air down against the floor provides absolute control of the warm air.

Correctly Engineered Unit Heaters by Airtherm

Capitalizing thirty years research and experimentation Airtherm Engineers have produced the ultimate in unit heater efficiency and design. In addition to the *AIRBLANKET*, the Airtherm Line includes the Airvector (Fan Driven Units) and the Airheator (Blower Driven Units)...both of which embody many new, exclusive features in unit heater construction and performance.

Write for full details of the Airtherm line including the *AIRBLANKET*, at once.

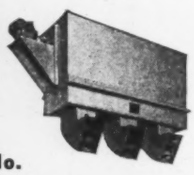
AIRTHERM MFG. CO.
1478 S. Vandeventer Ave., St. Louis, Mo.

Ten Years Success

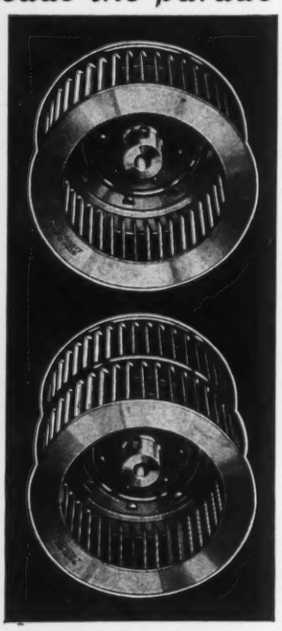
During the past ten years many of America's "Hardest-to-Heat" buildings have been comfortably and economically warmed by this simple, proven method of heat distribution . . . the Exclusive *AIRBLANKET* Method.

Don't buy any unit heater until you have secured full details on the *AIRBLANKET*.

The AIRBLANKET is available for ceiling or floor mounting.



TORRINGTON EQUIPMENT *leads the parade*



Backed by broad experience in the industry, TORRINGTON engineering is years ahead — is leading the way to increased dependability and better quality. Consider these points and what they can mean to your product.

- **Smoother, quieter operation.**
- **Non-rusting, non-resonant construction.**
- **Lower starting torque.**
- **Lighter in weight yet rugged.**
- **Less power consumption.**
- **Every wheel hand inspected and statically balanced.**

Write for information about our improved line of propeller fans. Also ask for our catalog of blower wheels with ratings for all sizes and table of dimensions for housing scroll design.

TORRINGTON MFG. CO. TORRINGTON CONN. U. S. A.
TORRINGTON FANS AND BLOWER WHEELS

Conventions and Expositions

Chicago Exposition of Power and Mechanical Engineering: October 4-9, International Amphitheatre, Chicago, Ill. Under management of the International Exposition Co., Grand Central Palace, New York, N. Y., manager of the biennial National Expositions of Power and Mechanical Engineering, the next of which will be held in New York City December 5-10, 1938.

American Public Health Association: Annual meeting, October 5-8, Hotel Pennsylvania, New York, N. Y. Headquarters Office, 450 7th Ave., New York, N. Y.

American Welding Society: Annual convention, week of October 18, Hotel Traymore, Atlantic City, N. J. Headquarters Office, 33 W. 39th St., New York, N. Y.

Metal and Welding Exposition: Week of October 18, Convention Hall, Atlantic City, N. J. American Society for Metals, 7016 Euclid Ave., Cleveland, Ohio.

Refrigeration Service Engineers' Society: Annual convention and exhibit, November 3-5, Hotel Stevens, Chicago, Ill. Chairman, Herman Goldberg, 9 S. Clinton St., Chicago, Ill.

American Petroleum Institute: Annual meeting, November 9-12, Hotel Stevens, Chicago, Ill. Headquarters Office, 50 W. 50th St., New York, N. Y.

International Acetylene Association: Annual convention, November 10-12, Hotel Tutwiler, Birmingham, Ala. Headquarters Office, 30 E. 42nd St., New York, N. Y.

Chemical Industries Exposition: December 6-11, Grand Central Palace, New York, N. Y.

Drying and Air Conditioning Symposium: December 27-28, University of Pennsylvania, Philadelphia, Pa. Under auspices of the Division of Industrial and Engineering Chemistry, *American Chemical Society*. Symposium Chairman, D. B. Keyes, Department of Chemistry, University of Illinois, Urbana, Ill.

Fifth International Heating and Ventilating Exposition: January 24-28, 1938, Grand Central Palace, New York, N. Y. Manager, Charles F. Roth, International Exposition Co., Grand Central Palace, New York, N. Y.

American Society of Heating and Ventilating Engineers: Annual meeting, fourth week in January, Biltmore Hotel, New York, N. Y. Secretary, A. V. Hutchinson, 51 Madison Ave., New York, N. Y.

National Warm Air Heating and Air Conditioning Association: Winter convention, January 24-26, Roosevelt Hotel, New York, N. Y. Managing Director, Allen W. Williams, 50 W. Broad St., Columbus, Ohio.

American Society of Refrigerating Engineers: Annual meeting, January 26-28, Roosevelt Hotel, New York, N. Y. Secretary, David L. Fiske, 37 W. 39th St., New York, N. Y.

Recent Trade Literature

For your convenience in obtaining copies of bulletins, see coupon on page 160.

No. 2493. AIR CONDITIONING: Carrier Corp., 850 Frelinghuysen Ave., Newark, N. J. 12 p. booklet on equipment for industrial air conditioning, refrigeration, and space heating showing pictures and descriptions of equipment ranging from a one room "Weathermaker" to a self contained centrifugal refrigerating machine for use on large jobs. A feature is an explanation of the conditions under which the various equipment best operates.

No. 2494. AIR CONDITIONING: Carrier Corp., 850 Frelinghuysen Ave., Newark, N. J. 4 p. pamphlet illustrating and describing two new summer air conditioning units, one a

Welding fittings at their best

WELD your pipe lines—for added safety and added strength with lighter weight—for smooth-flow interiors—for smooth, sightly, easily insulated exteriors. Taylor WeldELLS head the list of Taylor Forge *Engineered* Welding Fittings. They are reinforced at the crotch where strain is greatest—have tangents which remove the weld from the curved section—have machine tool bevelled welding surfaces which permit sound, dense welds.

The Taylor Forge line covers every need. It is the most complete line made.

TAYLOR FORGE & PIPE WORKS
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Pittsburgh Office: Gulf Bldg.

Also made from BYERS GENUINE WROUGHT IRON PIPE

TAYLOR FORGE
WeldELLS
SEAMLESS PIPE FITTINGS FOR WELDING

Bound For the MARTS OF AMERICA

Tremendous stacks of timber! Soon every piece will be transformed into paper or pulp, but millions of gallons of water must be used. And for a dependable supply of water at low cost, the world's greatest paper and pulp mills turn to Layne. Whatever your water needs may be, Layne and Bowler, Inc. and their many affiliated companies offer you the most dependable water producing experience, knowledge and equipment obtainable from any source.

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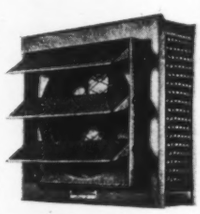
INTERNATIONAL WATER SUPPLY, LTD., FORT ERIE, N. . . . . ONTARIO, CANADA.

LAYNE & BOWLER, INC.
Dept. K, Memphis, Tenn.

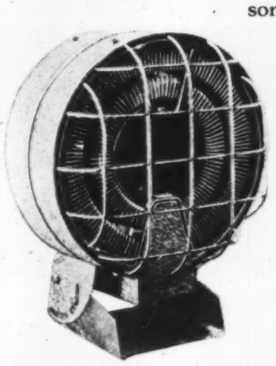
LAYNE PUMPS
WELL WATER SYSTEMS

Get set for Fall and Winter *with* CHROMALOX *ELECTRIC* UNIT HEATERS



Snappy days are coming. Are you going to meet occasional subnormal weather with the ponderous, expensive winter heating system? A few of these electric unit heaters will provide all the warmth needed for these Fall days at a fraction of the operating cost of the main heating plant.

And what about the dead of winter, when you just can't heat some spots in the plant? Again Chromalox electric unit heaters will serve nobly to aid the main heating system.

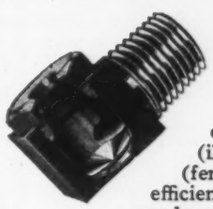


Overtime work, too, presents a further opportunity for saving. Shut down the boilers and snap on your Chromalox unit heaters in the departments still working. It's cheaper.

Made in types for office or shop, in capacities from 1.5 Kw. to 20 Kw. (5,118 to 68,240 B.t.u.) Learn all about them in the 64-page Chromalox catalog—on request.

EDWIN L. WIEGAND COMPANY
7610 Thomas Blvd. Pittsburgh, Pa.

MONARCH *non-clogging* SPRAY NOZZLES



For Air Washing

We recommend either ¼" or ⅜" Fig. 629 male pipe (illustrated) or ¼" Fig. 631 (female) Brass nozzles. Small, efficient, and inexpensive, they produce an evenly distributed hollow cone spray. One large lead hole minimizes any clogging tendency. Capacities from 4.7 G.P.H. up. Now being used by some of the largest air conditioning manufacturers.

For Re-Cooling

Monarch Fig. B-8 and B-8-A cast Brass nozzles, for cooling condensing water in spray ponds, use no internal vanes or deflectors which might facilitate clogging. Available in capacities up to 104 G.P.M. at 10 lbs.

Write for New Catalog 6-A

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2728 E. Westmoreland Street, Philadelphia, Pa.

BEFORE MERCURY STARTS TO SLIDE
MAKE A HEATING SYSTEM CHECK-UP
In a few weeks it will be too late to take full advantage of the savings that a thorough check up will provide. Also it will be nearing the time when complaints on both heating and hot water service are at the peak.
One move now will forestall both these losses. Usually a small investment produces a big saving. In any event, it costs nothing to find out how much you could save.
May we send our nearest representative to look over your heating and hot water situation, or make suggestions by mail?
GET THIS NEW DATA BOOK— with hook-ups, and calculation data for all types of heating, air conditioning, process equipment, and hot water control.
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REPRESENTATIVES IN ALL PRINCIPAL CITIES
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FROM LEAKS and BREAKDOWNS
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GRID UNIT
It is distinctive among unit heaters. The "fin" or heating sections are all cast aluminum bonded to cast iron steam chamber—no soldered, brazed or expanded joints—no pressed or rolled joints—and it cannot be affected by electrolysis to produce corrosion. Because of its sturdy construction it can neither be bent out of shape nor broken. It's permanent equipment that requires no maintenance and will last indefinitely. Complete details on request.
THE UNIT HEATER & COOLER CO., Wausau, Wisconsin
Offices in all principal cities
GRID
UNIT HEATER

floor model and one a ceiling model which use cold tap, or well water as a cooling medium.

No. 2495. AIR CONDITIONING: Fairbanks-Morse & Co., 900 S. Wabash Ave., Chicago, Ill. 32 p. bulletin on central system air conditioners, discussing and illustrating the various types of units, their functions, and their applications in office buildings, shops, industrial plants, residences, etc. The several models and sizes of the units for performing various functions are described in some detail, and specifications are given.

No. 2496. AIR CONDITIONING: Williams Oil-O-Matic Heating Corp., 1201 E. Bell, Bloomington, Ill. 20 p. fully illustrated booklet on summer and winter air conditioning by low pressure steam, describing the "Air-O-Matic" absorption refrigeration unit, its method of operation and applications. Data on the operating cycle, steam, electrical, and cooling water requirements are given and information is presented on control. Two color diagrams show the details of the evaporative cooled absorption unit and the water circulating absorption unit.

No. 2497. BLOWER WHEELS: Torrington Mfg. Co., 70 Franklin St., Torrington Conn. Loose-leaf catalog containing information on "Autocrat" paddle type blower wheels, aluminum alloy blower wheels, steel wheels, and numerous performance and capacity tables.

No. 2498. BOILERS: International Boiler Works Co., East Stroudsburg, Pa. 12 p. bulletin on "Fuel-Saver" water tube steel heating boilers for large capacity heating plants (oil burning, hand fired coal and stoker firing) with complete information on features and design. A knockdown type is featured.

No. 2499. BOILERS: National Radiator Corp., 221 Central Ave., Johnstown, Pa. 12 p. bulletin giving specifications for "multi-finger heat extractor" boilers for hand, stoker, and oil firing.

No. 2500. CENTRIFUGAL REFRIGERATION: Carrier Corp., 850 Frelinghuysen Ave., Newark, N. J. 48 p. semi-technical manual on centrifugal refrigeration treating of the principles and operation of the centrifugal machine, embodying the use of the refrigerant "Carrene." Adaptability of this type of apparatus for small and large tonnages, applications ranging from comfort cooling to industrial process and refrigeration work down to —110 F is described and its use in economical heat balance applications, and in combination with existing ammonia equipment, is discussed. Engineering data, a description of accessories and parts, and thermodynamic characteristics of refrigerants complete the booklet. An extensive list of installations is appended.

No. 2501. COMPRESSORS AND VACUUM PUMPS: Ingersoll-Rand, 11 Broadway, New York, N. Y. 32 p. catalog of "Type 30" industrial compressors and vacuum pumps, with information on selection of the proper compressor, types, regulation or control, and rating tables for the various models.

No. 2502. COMPRESSORS AND VACUUM PUMPS: Ingersoll-Rand, 11 Broadway, New York, N. Y. Folder describing air cooled compressors and vacuum pumps for industrial applications, ¼ to 10 hp, 30 in. vacuum to 1000 lb pressure.

No. 2503. CONTROLS AND SWITCHES: Paragon Electric Co., 37 W. Van Buren St., Chicago, Ill. 4 p. folder describing controls for air conditioning, stokers, oil burners, and refrigeration, including time switches, ventilation timers, and relays and defrosters. Also a 4 p. bulletin on time switches available with indoor or outdoor cases, describing features of their design and construction.

No. 2504. EXPANSION JOINTS: Yarnall-Waring Co., 102 E. Mermaid Lane, Philadelphia, Pa. 16 p. bulletin on expansion joints describing their features in detail and giving dimension tables, weights, prices, and describing features of construction. Particular attention is given to explaining the "Gun-Pakt" feature.

No. 2505. FANS: Kelvin-White Co., 90 State St., Boston, Mass. 12 p. booklet on "Torpedo" fans for ventilation work, describing their features and characteristics, the types available, and presenting an abridged table of sizes and capacities.

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Dull silver, trimmed with chrome, blends with prevailing white and pastel shades of the modern room, or the warm darker tones of the English type.

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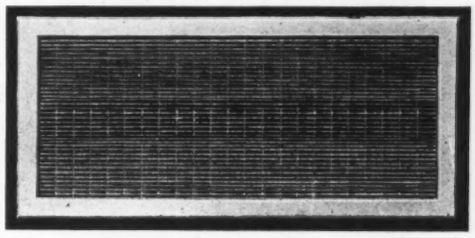

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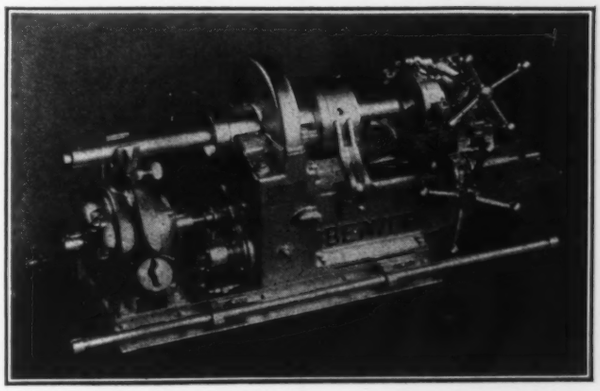

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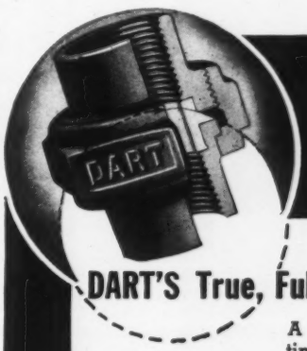

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Sales Agents, The Fairbanks Company, New York, and all branches. Canadian Factory: Dart Union Company Ltd., Toronto, Canada

No. 2506. GAS BURNERS: National Machine Works, People's Gas Bldg., Chicago, Ill. Loose-leaf booklet of bulletins on "Lo-Blast" gas conversion burners, with capacities ranging from 100,000 to 5,600,000 Btu per hr, describing features and giving specifications and instructions on installation in various types of boilers. Data on control are also presented.

No. 2507. HARD FACING: Haynes Stellite Co., 205 E. 42nd St., New York, N. Y. 104 p. book on hard facing with this company's products, describing some 500 money saving applications of the process, including examples of hard facing valves and the use of Stellite trim for high temperature, high pressure steam valves. New sections in this fourth printing of the booklet contain information on corrosion resistant "Hastelloy" alloys. Numerous illustrations are shown.

No. 2508. HEATING, AIR CONDITIONING AND VENTILATING: Melchior, Armstrong, Dessau Co., 300 Fourth Ave., New York, N. Y. 136 p. catalog on oil burner supplies, heating, air conditioning and ventilating equipment, including many new items, particularly in the air conditioning section. An extensive reference data section is included and several pages are devoted to hook-up diagrams for various types of controls.

No. 2509. HUMIDIFIERS: Rega Mfg. Co., 79 Mt. Hope Ave., Rochester, N. Y. Leaflet describing spray humidfiier units for steam, vapor, hot water, or warm air heated buildings.

No. 2510. INDUSTRIAL FINISHES: Roxalin Flexible Lacquer Co., Inc., Elizabeth, N. J. Data sheet summarizing the properties of a tarnish proof, cellulose type, air drying flexible lacquer, available for spraying and dipping. Also a data sheet for a brass and silver lacquer for outdoor exposure for resistance to wind and weather.

No. 2511. MONEL METAL: International Nickel Co., Inc., 67 Wall St., New York, N. Y. 16 p. technical data bulletin on the engineering properties of "K" Monel, including information on composition, physical constants, properties, mill and shop working, heat treatment, welding and soldering, corrosion resistance, etc. Among the applications illustrated are pump shafts, wearing sleeves for small centrifugal pumps, valve discs, seat rings, and ball bearing assemblies.

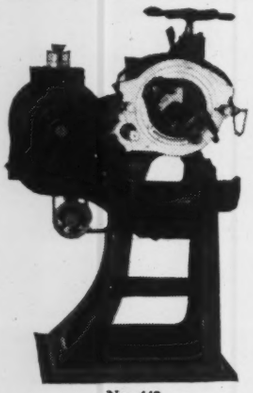
No. 2512. PIPE: American Rolling Mill Co., 703 Curtis St., Middletown, Ohio. 44 p. booklet on 6 to 36 in. pipe for industrial uses, listing the advantages of "Armco" spiral welded pipe, giving technical data on expansion and wall thickness, and presenting standard dimensions. Exterior, interior, and hot dip coatings applied to the pipe at the mill are described, and flow charts and information for field installations are included.

No. 2513. PULLEYS: Link-Belt Co., 307 N. Michigan Ave., Chicago, Ill. 8 p. booklet on cast iron pulleys for power transmission, giving pertinent data on solid and split, single and multiple machine moulded pulleys, with and without rubber lagging.

No. 2514. PUMPS: Fairbanks, Morse & Co., 900 S. Wabash Ave., Chicago, Ill. Two new bulletins on this company's line of deep well turbine pumps with enclosed impellers. One deals with oil lubricated units used for most applications, and the other covers water lubricated units used where the water must be kept pure and clean and oil lubrication is not permissible. Both types are available in sizes from 4 to 16 in. capacities from 10 to 2300 gpm, with lifts ranging to 400 ft.

No. 2515. PUMPS: Fairbanks, Morse & Co., 900 S. Wabash Ave., Chicago, Ill. Bulletin describing duplex steam pumps for boiler feed and general service, designed to pump water, oil, and similar liquids at pressures up to 420 lb per sq in. in quantities up to 464 gpm. Several refinements in design and construction are described.

No. 2516. PUMPS: Peerless Pump Div., Food Machinery Corp., 300 Walnut Rd., Massillon, Ohio. 64 p. catalog of deep well turbine pumps, with a detailed and fully illustrated description of the features of this company's equipment, manufacturing facilities, and testing department. Information and lists on applications are given, including use of the pumps in air conditioning work. A section of engineering data is included.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LESS POWER TO OPERATE!

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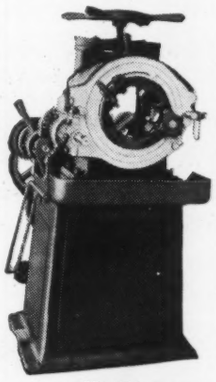
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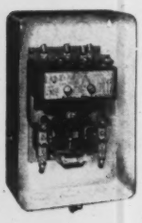

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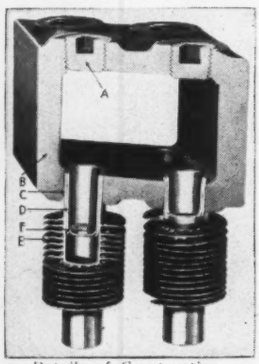

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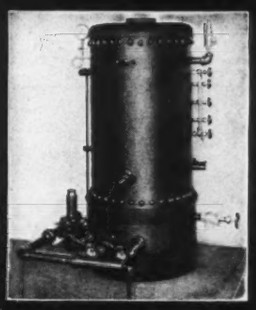

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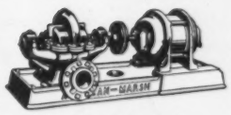

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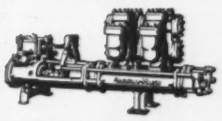

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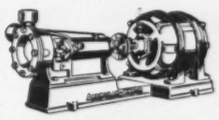

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And so to better associate our Company name with our long established trade name — American-Marsh — by which most of our equipment is known, we have recently changed our Company name to that used in the signature below.

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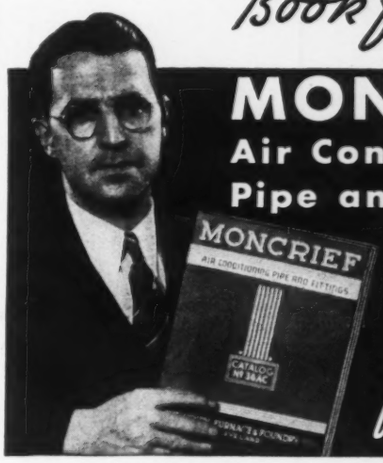

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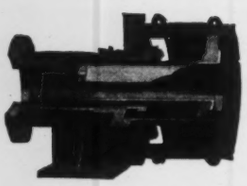

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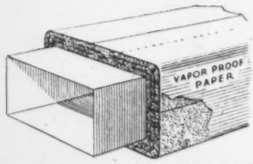

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2521	2522	2523	2524	2525	2526	

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No. 2517. PUMPS: Roots-Connersville Blower Corp., 16th St. & Columbia Ave., Connersville, Ind. 6 p. bulletin on turbine pumps for the industrial field, illustrating and describing constructional features and giving a selection table.

No. 2518. SHEARS: Niagara Machine & Tool Works, 637 Northland Ave., Buffalo, N. Y. 8 p. bulletin on power squaring shears built in 6 to 12 ft cutting lengths with 10 ga capacity, listing features and giving specifications.

No. 2519. SPRAY NOZZLES: Monarch Mfg. Works, Inc., Salmon & Westmoreland Sts., Philadelphia, Pa. 8 p. bulletin on spray nozzles for air washing and air conditioning, describing in detail the features of the various types and giving capacity tables.

No. 2520. STOKERS: Winkler Mfg. Corp., Lebanon, Ind. Series of 4 p. bulletins giving information on various models of stokers for bituminous coal, describing features of construction, giving specifications, dimension drawings, and information on installation.

No. 2521. TOOLS: J. H. Williams & Co., 75 Spring St., New York, N. Y. 184 p. pocket size booklet entitled "Tools of Industry" describing this company's entire industrial line of drop forged wrenches and other standard stock specialties. Many new lines are listed and described.

No. 2522. UNIT HEATERS: John J. Nesbitt, Inc., State Rd. & Rhawn St., Philadelphia, Pa. 36 p. catalog describing and illustrating the new de luxe design propeller fan unit heaters manufactured by this company and distributed through Warren Webster & Co. Engineering data are included.

No. 2523. VALVES: General Electric Co., 1 River Rd., Schenectady, N. Y. Data sheet on "Thrustor" operated valves for pipe sizes 1 to 10 in., illustrating and describing features and giving dimensions.

No. 2524. VENTING VALVES: Gorton Heating Corp., Cranford, N. J. 4 p. bulletin describing equalizing valves for even and quick heating of radiators and eliminators for the quick venting of steam mains.

No. 2525. WATER CIRCULATORS: General Blower Co., Vineland, N. J. 4 p. bulletin on "G. B. C." water circulators for hot water heating plants, describing features, giving sizes, capacities, dimensions, and installation data.

No. 2526. WELDING PIPE LINES: Lincoln Electric Co., 12818 Coit Rd., Cleveland, Ohio. 14 p. bulletin entitled "Building Better Pipe Lines Faster at Less Cost" by shielded arc welding, giving information on construction of pipe lines for the transportation of oil, gas, and water. In addition to describing the advantages of this process, and illustrating it, sample specifications for the arc welding of oil and gas pipe lines, and for the arc welding of steel water piping are presented.